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PROPANE/DIESEL INJECTION SYSTEM FOR DIESEL TRUCK ENGINES

Prepared for the

TRANSPORTATION DEVELOPMENT CENTRE

by

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The contents of this report reflect the views of the contractor and not necessarily the official views or policies of the Transport Canada Research and Development Centre.

Un sommaire en français de ce rapport est inclus avant la table des matières.

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17. Abstract

Propane is commonly used as a fuel for vehicles owing to its low cost, good performance, and low emission characteristics. Propane is usually introduced into the engine intake manifold at low pressure, a process known as fumigation. Such a technique when applied to diesel engines can result in poor ignition and reduced engine life. The present study researched the injection of liquid propane at high pressure directly into the combustion chamber of a diesel engine, with ignition provided by a pilot injection of No. 2 diesel fuel. The results, obtained using a single-cylinder engine, indicate that good engine thermal performance and greatly reduced smoke emissions can be achieved. Economic studies show that a developed liquid 2-fuel injection system would be of interest to truck fleet operators in several regions of Canada.

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17. Résumé

Le propane est souvent utilisé comme carburant pour véhicules en raison de son prix avantageux, sa performance souple, et son niveau d'émanations réduites. Le propane est normalement alimenté à basse pression dans le moteur à travers le collecteur d'admission, un procédé qu'on appelle l'aspiration. Cette technique pourrait s'avérer négative dans le cas des moteurs diesels et aurait pour résultat un allumage incomplet et une durée de vie réduite du moteur. Cette étude de développement portait sur l'injection du propane liquide à haute pression introduit directement dans la chambre de combustion d'un moteur diesel dont l'allumage était déclenché par une injection pilote de carburant diesel No. 2. Les résultats obtenus sur un moteur monocylindre indiquent un bon rendement thermique et un niveau trés réduit d'émanations de fumée. L'étude économique démontre qu'un système d'injection à deux carburants liquides pourrait être intéressant pour les propriétaires de flottes de camions dans plusieurs régions du Canada.

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SUMMARY

This report presents the results of a test program sponsored by the Transportation Development Centre (TDC) to investigate the use of propane in diesel engines utilizing the technique of direct liquid injection of propane with ignition provided by a pilot injection of No. 2 diesel fuel.

The objectives of the program included:

- determining the performance of a laboratory engine operating in two modes of propane fuelling, i.e. fumigation and also liquid injection directly into the combustion chamber, with diesel pilot ignition in both cases;

- assessing the potential of a dual-fuel liquid propane injection system as an alternative fuel technology, based upon laboratory test results using prototype hardware.

The engine test work was conducted on a single-cylinder laboratory engine having a pre-chamber combustion system, employing twin plunger-type fuel injection pumps mounted inline and connected by variable-phase angle couplings to permit variations in timing of both fuel injection pulses. Tests were conducted using separate twin injectors and also injectors capable of introducing both propane and pilot diesel fuel through a single body fitted to the normal injector cylinder head aperture.

The results of the testing have shown that liquid propane could be successfully pumped, injected and ignited in the test engine, and that the energy conversion performance was close to that of a diesel fuel system. The indications are that with optimization of the prototype system tested, engine operation would be as efficient as that achieved with developed diesel fuel systems.

SUMMARY (Cont'd)

Both gas-fumigated and liquid injected propane test results show large reductions in exhaust smoke emissions as compared to operation on diesel fuel, a situation having special implications regarding diesel engines in urban service.

Economic studies of the projected fuel cost savings using a dual-fuel liquid propane injection system applied to a large diesel truck indicate that a developed system would be of interest to truck fleet operators in regional areas such as Toronto, Edmonton, and Vancouver.

SYSTÈME D'INJECTION PROPANE/DIESEL
POUR MOTEURS DE CAMIONS DIESELS

SOMMAIRE

Ce rapport présente les résultats d'un programme d'essais pour Transports Canada dans le but de déterminer les caractéristiques d'allumage d'un moteur diésel alimenté en propane avec injection de carburant diésel no. 2, en utilisant un injecteur capable de propulser les deux carburants dans une ouverture de culasse standard.

Le programme avait comme objectifs:

- de déterminer les performances du moteur lorsqu'il opère en mode "double carburant" (propane injecté avec allumage pilote au diésel) comparativement à la performance utilisant de l'huile diésel no. 2 seulement.

- de déterminer des critères d'équipement pour un système à injection grandeur réelle capable de manipuler du propane liquide et de l'huile diésel no. 2 (en utilisant le principe d'allumage pilote).

Le programme d'essai du moteur a été effectué en utilisant un moteur monocylindre de laboratoire. Des essais ont d'abord été effectués pour établir les critères de performance du moteur utilisant du carburant diésel no. 2 à l'aide du moteur standard. Les essais de référence ont été effectués sur un dynamomètre à trois rapports de compression et à trois vitesses de moteur. Le couple du moteur, la consommation en carburant, l'opacité de la fumée évacuée, le niveau d'oxyde de carbone, les hydrocarbures et les émissions d'oxyde d'azote ont été calculés à partir des données recueillies.

SOMMAIRE (Suite)

Les résultats du design, du développement et des essais ont démontré que le propane peut être injecté sous forme liquide dans un moteur diésel qui pourra alors opérer d'une façon efficace comme s'il opérait uniquement avec du carburant diésel.

Les principes développés dans le système d'injection double propane/diésel no. 2 peuvent s'appliquer à d'autres combinaisons de combustibles comme diésel non normé no. 2 ou méthanol (diésel no. 2).

La viabilité d'un système d'injection double carburants propane/diésel no. 2 a été démontrée. Les performances d'un tel système se comparent favorablement aux performances de base d'un système diésel. Finalement, l'émission de particules solides a été radicalement réduite.

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- RD-4-020 Injector
- SK-85111901 Injector Machining and Assembly

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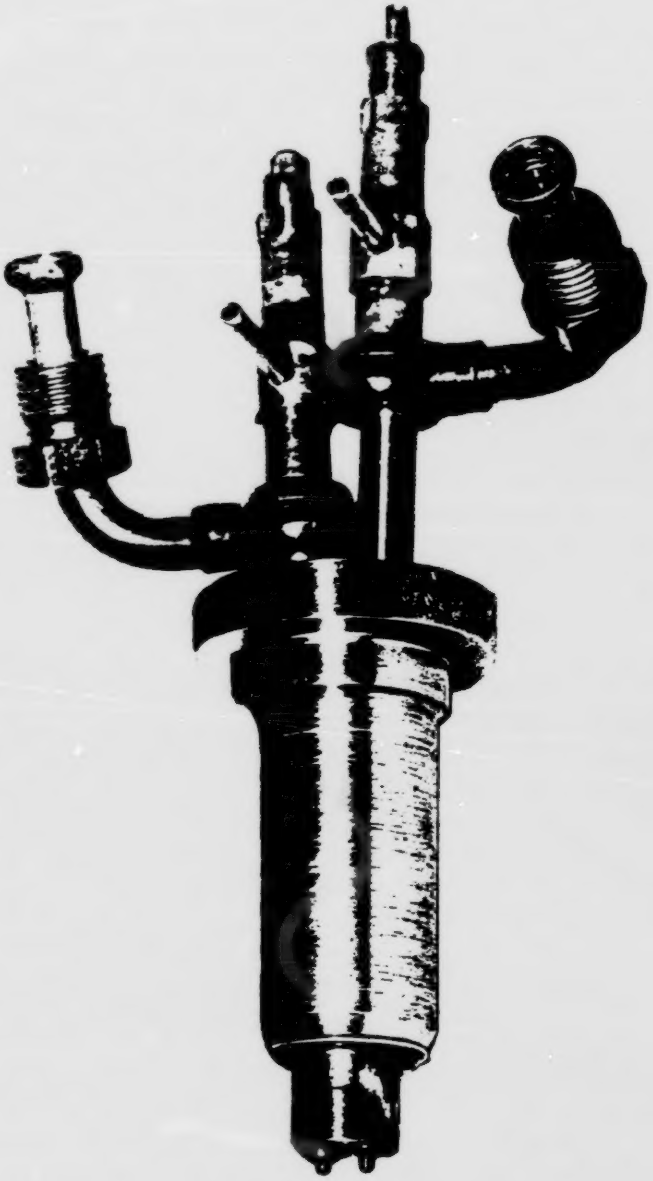

2- Fuel Injector For Liquid Propane and Pilot Diesel Fuel

1.0 INTRODUCTION

1.0 INTRODUCTION

Although propane has been accepted as an established fuel alternative to gasoline for Otto cycle engines, it has not received similar acceptance as a fuel for diesel engines. This is due in part to its low cetane number which necessitates some form of ignition assist.

Several techniques are available to accomplish propane ignition in diesel engines. The approach most commonly used has been to treat propane as an alternative fuel to displace a portion of the diesel fuel and use the remaining diesel fuel as a pilot ignition source. A simple engine modification can then be made by the addition of a propane tank and low pressure supply line, with the propane metered in gaseous form into the engine manifold - commonly referred to as "fumigation".

Unfortunately this simple "strap-on" modification concept has major drawbacks. The fumigation approach means that the quantity of air drawn into the engine cylinders with each stroke is reduced by the volume of propane introduced, thus the power output of the engine is inherently reduced (this characteristic should not be compared with fumigation systems used to increase power output of diesels wherein cylinder pressure restrictions are relaxed). A more serious problem, however, concerns the combustion characteristics of the fumigated gas/air mixture which in practice limit the amount of diesel fuel which can be displaced to only about 35% for most commercial vehicle installations.

The concept of direct injection of propane is attractive since in place of a fumigated gas/air charge in the cylinder, liquid propane is injected directly into the cylinder at the appropriate time in a controlled manner. Initial studies had shown that despite this promise, very little work had been done in this area, virtually no data was available on actual experience of handling propane at diesel fuel injection system pressures using existing diesel fuel system components, and no experimental engine test data was available.

Since two fuels are considered for injection but the cost of development of a new engine might not be warranted, it follows that an ideal system would permit both fuels to be injected through the same cylinder head injector aperture. In this way the high cost of cylinder head development for either new or retrofit applications would be eliminated.

It was in the foregoing context that the present dual-fuel injector concept and associated engine test program was developed.

In following sections of this report the organization of work, injector development considerations and the engine test program are presented and discussed.

2.0 OBJECTIVES

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2.1 Overall Objectives

- To reduce Canada's dependence on imported oil-based fuels by displacing oil consumed by diesel engines in road transportation with propane.
- To develop equipment for the direct injection of propane and a "pilot" quantity of diesel oil into a diesel truck engine, suitable for retrofit and new applications.

2.2 Project Objectives

- To determine the performance of a Waukesha CFR 48 single cylinder pre-chamber engine operating on:

  a) Diesel fuel only
  b) Injected propane and diesel pilot ignition
  c) Fumigated propane and diesel pilot ignition

  and to compare the results under steady state conditions.

- To assess the potential of a propane dual-fuel injection system as an alternative vehicle fuel technology.

3.0 PROJECT ORGANIZATION

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3.1 Project Activities

Major individual project activities included preparation of draft and contractual work agreements; current technology review; baseline diesel engine testing; propane fumigation engine testing; 2-fuel pump procurement and checkout testing; 2-fuel injector design, fabrication and bench testing; 2-fuel injector engine testing; manufacturing cost evaluation and economic analysis.

Control documentation for the project included a SPEC planning network developed at the outset of the work, in which the interrelated nature of the various activities was conveniently displayed (refer to Appendix C, "SPEC NETWORK DIAGRAM").

3.2 Project Team

The project team members included SNC Inc., the University of Toronto's Department of Mechanical Engineering (UTME), IVI Inc. (an SNC subsidiary located in Quebec City), and Southwest Research Institute (SWRI) located in San Antonio, Texas. Various specialist subcontractors were engaged as required during execution of the work, the most important of which was the Diesel Systems Group of Stanadyne Inc. located in Hartford, Connecticut.

3.3 Responsibilities

SNC was prime contractor, performing overall project management functions and being responsible for the specific tasks of technology review, basic injection system engineering, dual-fuel injector prototype design, test hardware coordination, preliminary market assessment, economic analysis and report preparation.

UTME was responsible for the conduct of a series of laboratory dynamometer engine tests using a dual-fuel pump and injectors supplied by SNC, followed by compilation and evaluation of test results.

IVI responsibilities included fabrication and bench testing of 2-fuel injectors using the motored dual-fuel pump, injector modification, and establishing tooling and manufacturing costs for the 2-fuel injector on a high volume basis.

The dual-fuel pump to supply separate high pressure streams of liquid propane and diesel oil was designed, built and checkout tested by SWRI, who also provided technical assistance to the project in areas of fuel spray and basic diesel engine combustion technology.

Stanadyne furnished prototype injector nozzle tips and also provided key technical guidance with regards to injector fabrication methods and associated equipment, stock parts identification, and assistance in locating design information.

A listing of key project team personnel is contained in the following section.

3.4 Team Key Personnel

SNC Inc. Montréal, Qué.	F.W. Eyre	Project Manager
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	L. Segal	Co-investigator
	U. Oester	Co-investigator
IVI Inc. Québec, Qué.	J.-P. Drolet	V-Pres. R&D
	S. Roy	Design Engineer
	M. St-Laurent	Development Specialist
SWRI San Antonio, Texas	R. Thring	Manager, Engine R&D
	R. Phatak	Engine Design Group Leader
Stanadyne Inc. Hartford, Conn.	B. Wysong	Dir. Product Engineering
	L. Tolan	Assist. Chief Engineer

4.0 2-FUEL INJECTOR DEVELOPMENT

4.0 2-FUEL INJECTOR DEVELOPMENT

4.1 Propane Characteristics as a Diesel Engine Fuel

"Propane" as commonly sold is composed of a mixture of hydrocarbons, chiefly propane, propylene and butanes. Known throughout the world as LPG (liquified petroleum gas), autogas, and also commercial propane, it is marketed for automotive use in North America as Grade I (or HD-5) propane for which a limit on the maximum level of propylene has been established.

Propane offers initial attractions as an alternative fuel by virtue of its wide availability through an established and expanding distribution network, its forecast continued domestic supply, its long history of commercial use, its known clean engine performance, and its low price on a thermal basis.

Being a fuel with a relatively high vapor pressure, propane is distributed and stored as a liquid under pressure. All design aspects of handling equipment are fully controlled by appropriate codes, including those for onboard vehicle installations.

Table 1 lists the most important properties of propane as an engine fuel, along with corresponding No. 2 diesel values for comparison.

It can be seen that the vapor pressure of propane increases rapidly with temperature, reaching a value of 4140 kPa at a typical underhood engine compartment temperature of 90°C. Accordingly, cavitation problems are to be expected when pumping propane at these temperatures, a situation that does not exist in the case of diesel oil.

TABLE 1 - FUEL PROPERTIES COMPARISON*

		Propane	No.2 Diesel
Boiling point at 1 atmos., °C		-43	150 - 360
Vapor pressure, kPa at	0°C	350	-
	30°C	725	-
	60°C	2000	-
	90°C	4140	1.0 - 3.0
Critical pressure, kPa		4250	-
Critical temperature, °C		97	-
Specific heat, liquid, 16°C		0.59	0.40 - 0.50
Specific gravity, gaseous, 16°C (air = 1.0)		1.55	-
Specific gravity, liquid at	0°C	0.57	0.86
	20°C	0.50	0.85
	40°C	0.47	0.84
	60°C	0.43	0.82
	80°C	0.38	0.81
Net calorific value, MJ/kg		46.3	44.2
Auto-ignition temp. in air, °C		490	220
Air requirement, kg/kg		15.5	14.5
Viscosity, centipoises		Refer to Fig. 13	

*derived from diverse sources

The thermal coefficient of expansion of propane is also extremely high, which affects its density and volumetric energy content and therefore dictates the use of a form of fuel delivery rate control with substantial temperature compensation. Table 1 indicates a 25% reduction in specific gravity between the temperatures of 0°C and 60°C, representing the range of fuel temperature variation during a typical winter engine warmup session. Diesel oil systems require less temperature compensation, having only a 5% reduction in S.G. over the same range of temperature.

Similarly, storage tanks must be designed to cope with propane's high expansion coefficient to avoid excessive tank wall tensile forces. Tank filling systems are constructed to automatically prevent more than 80% filling of total capacity, thereby trapping a cushioning volume of gaseous propane and air in the tank to absorb any subsequent thermal expansion of the liquid propane.

Propane has a higher mass calorific value and a lower density than No. 2 diesel fuel. If used as an injected compression ignition fuel, at a temperature of 20°C, approximately 60% more propane volume must be delivered from each injection pump stroke compared to No. 2 diesel oil to maintain the same engine power level. Substantial pump plunger/barrel modifications or complete pump replacement could be required to accomplish this in a converted engine.

In addition, supplementary pump lubrication is necessary to deal with propane's poor properties as a lubricant (termed "lubricity"), due principally to its low viscosity. For example, at 60°C the viscosity of kerosene (a liquid physically similar to No. 2 diesel oil) is 25 times that of propane.

As a substitute for diesel fuel, the consequences of propane's lower viscosity include reduced pumping power requirements (if delivery volume is unchanged), lower peak pump pressure requirements for equivalent spray characteristics, and a greater tendency towards leakage throughout the pumping circuit.

The auto-ignition temperature of propane is considerably higher at 490°C than that of No. 2 diesel oil at 220°C. The difference, in fact, is so great that under most conditions of operation propane will not ignite in a diesel engine without the provision of some form of ignition assistance. Sparks, glow plugs, plasma jets, chemical ignition additives and the presence of conventional diesel fuels are among the various alternative means available to achieve the satisfactory compression ignition of non-diesel fuels such as propane.

4.2 Technical Overview of 2-Fuel Systems

The term "dual fuelling" when applied to diesel engines generally implies the use of one of the following fuel feed arrangements:

- a combustible gas fumigated into the inlet air stream and ignited by a small "pilot" injection of good quality diesel fuel introduced through the normal injector.
- continuously or batch blended combinations of liquid non-diesel fuel and diesel oil introduced directly into the combustion chamber through a single injector.
- a primary non-diesel liquid fuel sprayed at low pressure into the inlet air stream and ignited by a small "pilot" injection of good quality diesel fuel introduced through the normal injector.

- a primary non-diesel liquid fuel and pilot diesel fuel injected directly into the combustion chamber through a common injector or through two separate injectors.

In fumigated systems the non-diesel fuel is metered into the inlet manifold as a gas at atmospheric pressure. Fumigated gas systems (natural gas and propane) are in popular use on a variety of diesel engines. Gaseous fuel compression-ignition is characterized by long ignition delay and high rates of cylinder pressure rise (knock), in addition to a tendency to misfire under light engine loads (high air/fuel ratios)(1). The volumetric efficiency and effective compression ratio are adversely affected by the fumigated gas, due to displacement of the intake air charge. Control hardware linking the gas metering device with the diesel fuel injection pump delivery rack adjustment is widely available for certain types of diesel engines. The main performance drawback concerns misfire at low loads, making necessary the use of anti-backfire valves, good fumigation metering systems, and high levels of pilot diesel fuel. Propane fumigation systems installed on diesel vehicles are currently limited to approximately 35% overall displacement of diesel oil, largely due to engine durability problems linked to high cylinder pressure peaks generated by the combustion of gaseous propane.

Blended dual-fuel systems are commonly used in stationary, railroad and marine engine applications where fuel stocks proportioning or onboard metering/blending systems can be adequately supervised. Research on high speed engines (2) has shown that the potential for displacement of diesel oils by non-diesel fuels using this approach is limited to relatively low percentages, probably due to unstable ignition caused by dilution of the pilot fuel charge, and that poor cold starting and fuel separation after blending remain practical obstacles. Measures taken to increase combustion chamber temperatures (inlet air heating, increased compression ratio and/or turbocharger boost) are known methods to improve combustion and increase the range of misfire-free engine operation (3) on blended fuels.

The technique of spraying liquid primary fuel into the inlet manifold airstream (commonly known as port injection) employs hardware commercially available for spark ignition gasoline fuel injection engines. Frequently used for alternative fuels dual-fuel testing, this approach is convenient for retrofit conversions at low capital cost. Due to the uniform distribution of the primary non-diesel fuel charge throughout the combustion chamber, homogeneous air/fuel mixtures are achieved (lack of charge stratification) leading to poor combustion behavior similar to that experienced when fumigating primary fuels. Inlet air heating and increased compression ratio are frequently required to achieve satisfactory performance levels. (4) Engine test work conducted using liquid propane sprayed into the inlet manifold produced similar findings, while demonstrating that solid particulate emissions could be greatly reduced using this technique of propane/diesel dual fuelling. (5).

Dual-fuel injection directly into the combustion chambers is an experimentally-established technique for realizing high levels of diesel oil displacement in high speed engines with few performance tradeoffs (2, 6). Since fuel charge stratification can be maintained for both fuels, correct air/fuel ratios can be achieved in the vicinity of the flame front generated by the pilot diesel spray. Moreover, care in control of injector spray orientation and projection through proper injection system calibration can result in good engine performance over the full range of speed and load. A commercial barrier to acceptance of this dual-fuel approach concerns the difficulty in injecting two fuels directly into the combustion chamber in two separate fuel streams. Experimental systems intended to prove the performance potential of this technique have employed two separate fuel pumps feeding separate injectors installed in highly modified cylinder heads. Others have placed the second injector in the aperture intended for the glow plug, with additional means provided for cold starting. The 2-fuel system conceived during the course of the present project will be described later in this report.

4.3 Selected Examples of Experimental Systems

Although several manufacturers offer commercial versions of 2-fuel injection systems for their low speed (mainly marine service) engines, a review of trade catalogues indicates that such systems for high speed engines are currently reserved for experimental and demonstration programs only.

A large number of patents are held by both manufacturers and individuals for 2-fuel injectors and related advanced injection systems. A review of recent patents in this area, with appropriate comments is contained in Appendix B. Important 2-fuel hardware developers include Caterpillar Tractor Co., Curtiss-Wright Corp., Lucas Industries Ltd., MWM AG, Robert Bosch GMBH, and Diesel Kiki Co.

The injection systems described below serve to illustrate the various approaches taken by other investigators working in this field:

o injector to deliver two different fuels through a common body, developed by the Mechanical Engineering Laboratory, MITI (Japan). In this design, co-axial nozzles separate the two fuel streams, allowing independent timing and delivery of the two fuel sprays. External dimensions of the injector body permit retrofit without cylinder head modifications (refer to Fig. 1).

- injector to deliver two different fuels through a common body, patented by G.C. Kriek (South Africa). Two fuel pumps are required, and the injector is of dimensions suitable for retrofit without cylinder head modifications. Both fuel sprays are delivered through the same orifices, consequently, partial mixing of the fuels will occur before injection, reducing the effectiveness of the pilot ignition fuel spray (refer to Fig. 2).

- injector to deliver two different fuels through a common body, patented by Klockner-Humboldt-Deutz AG (Germany). The design, with the addition of a check valve, appears to be identical to that of the Kriek injector described above (refer to Fig. 3).

- 2-injector system to accomplish pilot ignition of low grade alternative fuels in a locomotive engine, with fuel spray geometry patented by Southwest Research Institute (USA). In this case, the cylinder head is reworked by installing a pilot fuel injector in the pressure relief (blowdown) port. The existing unit injector has been retained for primary fuel injection purposes (refer to Fig. 4).

- 2-injector system to operate on diesel fuel oil as the primary fuel, and a chlorinated hydrocarbon waste liquid as the secondary fuel, patented by Dow Chemical Co. This system was developed for the purpose of waste disposal using

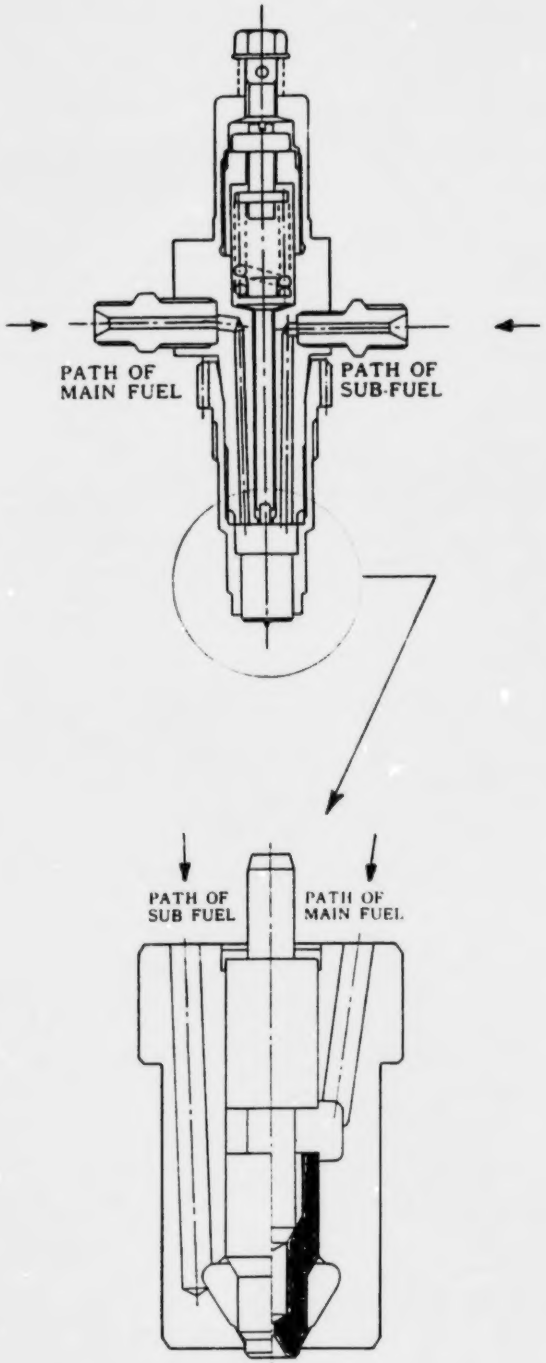
PATH OF MAIN FUEL
PATH OF SUB-FUEL
PATH OF SUB FUEL
PATH OF MAIN FUEL

Fig. 1 MITI 2-Fuel Injector

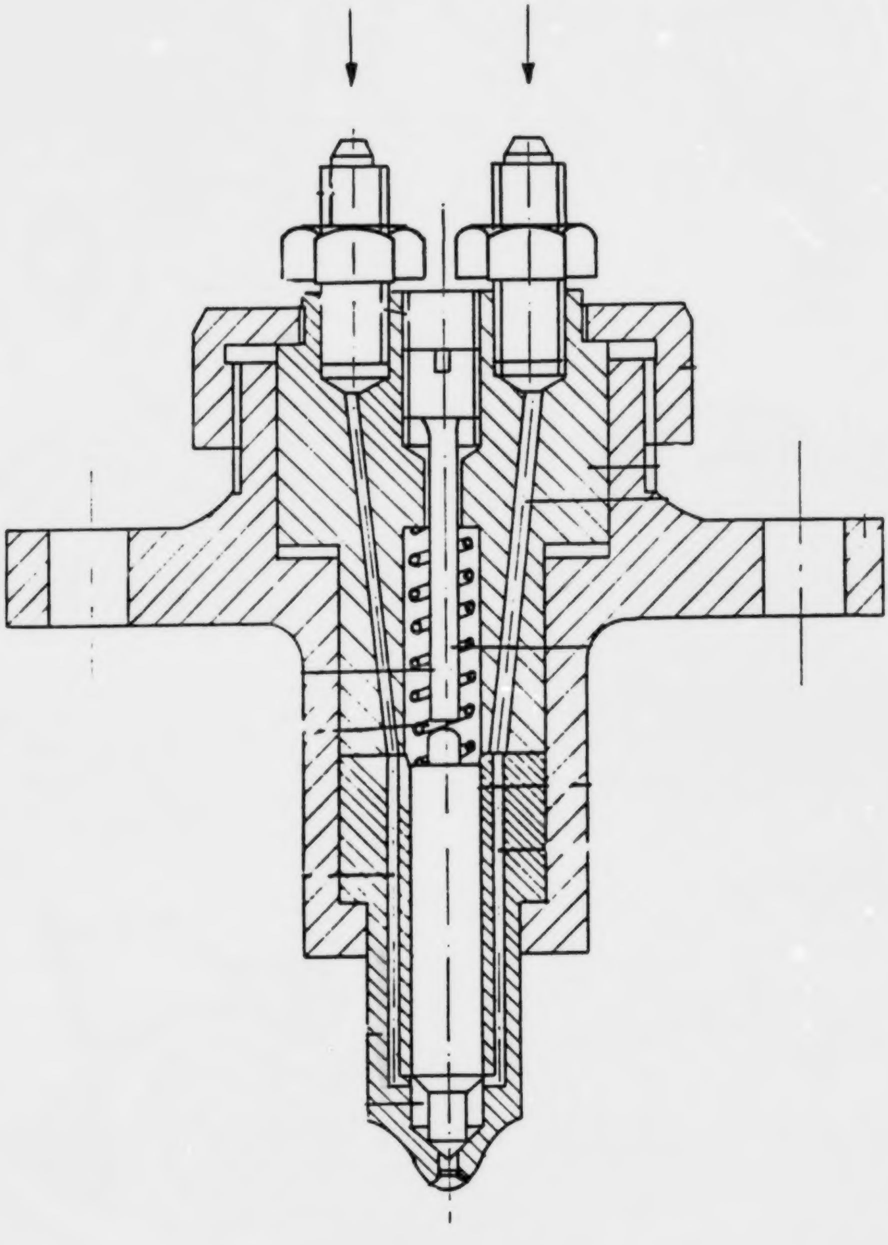

Fig. 2 G.C. Kriek 2-Fuel Injector

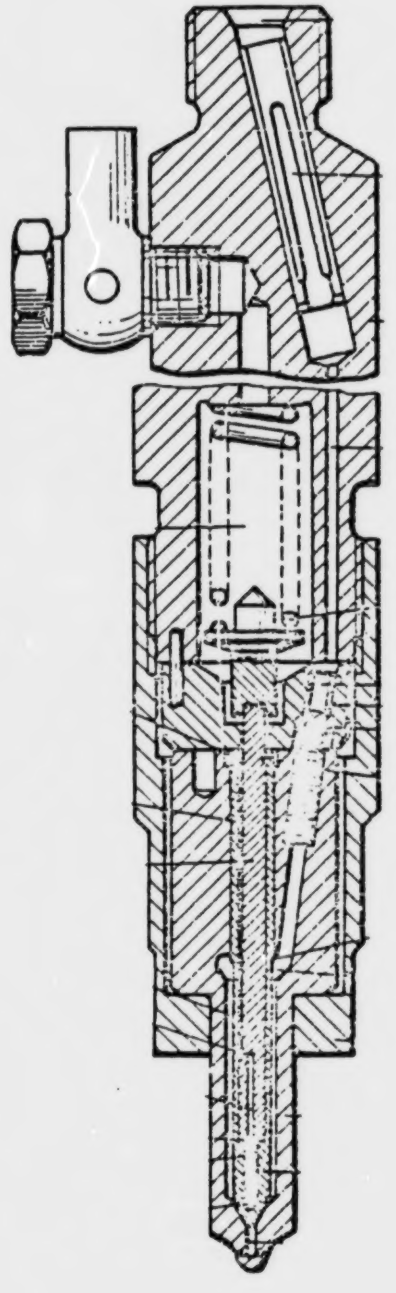

Fig. 3 KHD 2-Fuel Injector

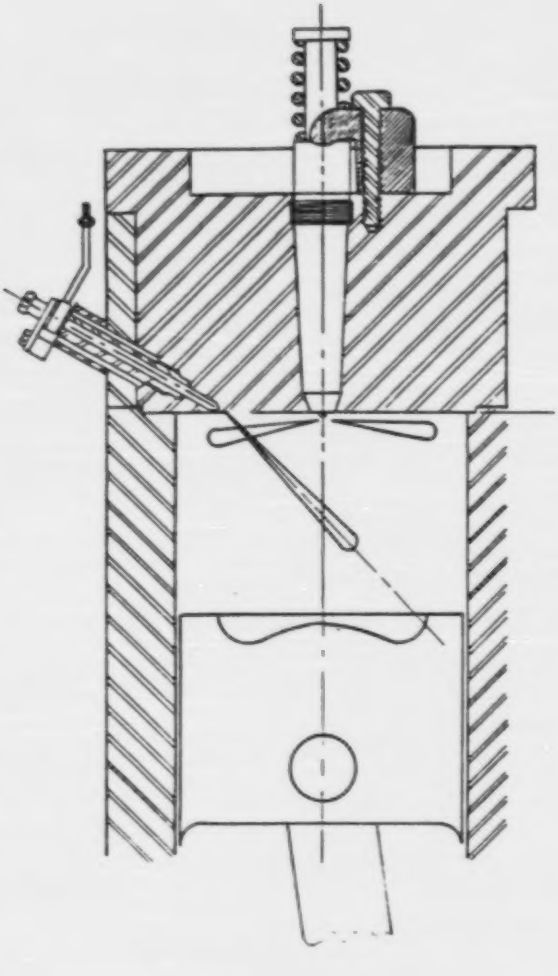

Fig. 4 SWRI 2-Injector System

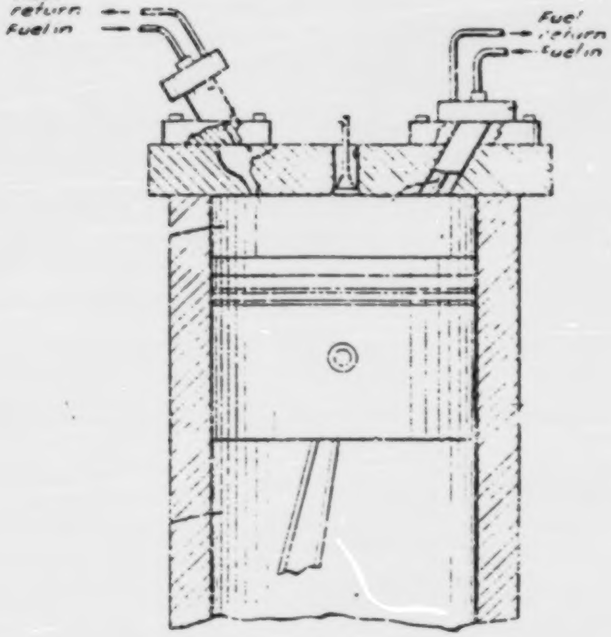
Fuel return
Fuel in
Fuel return
Fuel in

Fig. 5 Dow Chemical 2-Injector System

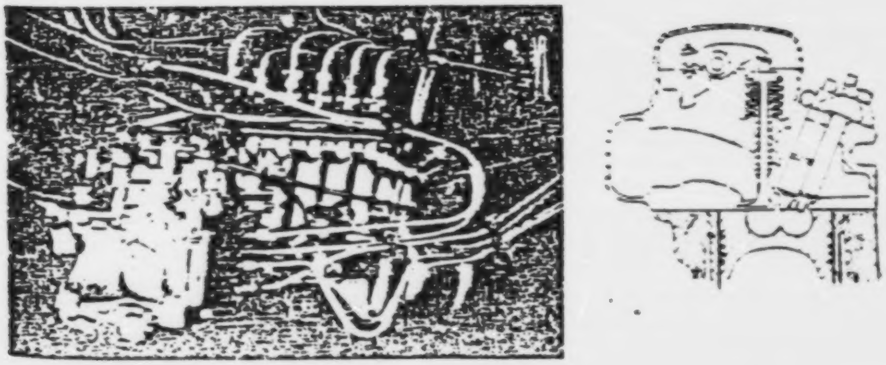

Fig. 6 Volvo 2-Injector System

cost-effective means. A possible advantage of this approach over blended fuel/waste mixture combustion methods is the potential ability to consume a higher proportion of waste liquid in the engine (refer to Fig. 5).

- 2-injector system to supply methanol or ethanol to a vehicle diesel engine, with ignition provided by a pilot injection of diesel oil. The system, built and tested by Volvo Truck Corp. (Sweden), requires extensive cylinder head modifications to install two separate injectors, and employs two conventional fuel injection pumps. (Refer to Fig. 6).

4.4 Fuel Injector Basic Principles

The output and efficiency of fuel injection engines are greatly dependent on the characteristics of the fuel spray injected from the nozzle into the combustion chamber. The process of spraying accomplishes two objectives: it divides the liquid fuel into a multiplicity of fine droplets to increase its surface area for more rapid heat transfer and combustion, and it distributes the fuel through the combustion chamber for intimate mixing of the fuel and air.

Spray Formation

Depending upon the injection pressure and viscosity of the liquid fuel, the first portion of fuel forced through the nozzle usually appears as a cylindrical jet. At a certain distance from the nozzle the jet breaks up into fine droplets to form a cone-shaped spray.

Several theories have been proposed to explain the process by which the solid stream of fuel issuing from a nozzle is broken up to form a spray. On the basis of present knowledge this process may be considered to take place in three steps:

1. Ruffling or disturbance of the jet surface from instability caused by such factors as surface tension, inertia forces, orifice imperfections, nozzle vibration, jet turbulence, and effervescence of dissolved gases.

2. Formation of ligaments and films as promoted by action of air friction on the initial disturbances of the liquid jet with subsequent break-up into fragments and then into spherical droplets from surface tension.

3. Further division into smaller droplets as long as the droplets have sufficient velocity with respect to the air.

The most widely used method for producing fuel sprays in diesel engines is by high pressure injection. Fuel is forced through one or more nozzle orifices under pressure to form unstable jets. The jets disintegrate upon leaving the nozzle and making contact with the air in the combustion chamber.

The uniformity of the spray, defined as the variation in droplet size throughout the spray, is a requirement which differs for various applications. In diesel engines, combustion should be progressive to avoid excessive firing pressures generated by simultaneous ignition of all the droplets in the spray. Therefore, the spray should consist of some fine droplets to initiate combustion and larger droplets for subsequent combustion. A graph of typical droplet size distribution as a function of injection spray pressure is illustrated in Fig. 7.

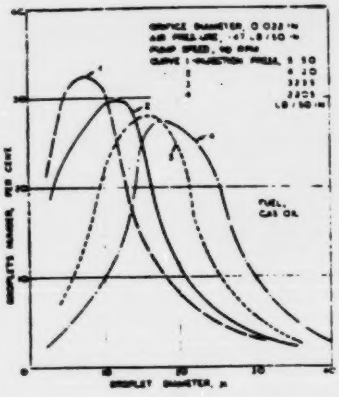

Fig. 7 Droplet Size Distribution Versus Injection Spray Pressure

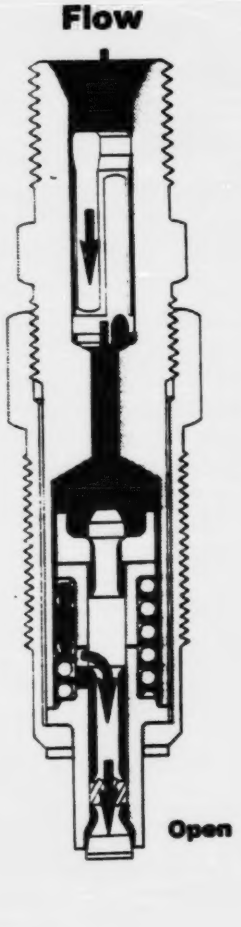

Fig. 8 Typical Open Injector Design

TABLE 2 FACTORS AFFECTING SPRAY CHARACTERISTICS

	Change	Break-Up Distance	Penetration (effective)	Dispersion	Droplet Size
Nozzle					
Orifice Diameter	increase	increase	increase	decrease	increase
Orifice L/D	various	optimum 1-2:1	minimum 1-2:1	optimum 4-2:1	optimum 4-6:1
Orifice Smoothness	rougher	decrease	decrease	increase	decrease
Opening Pressure	increase	decrease	increase	increase	decrease
Operating Conditions					
Injection pressure	increase	decrease	increase	increase	decrease
Discharge quantity	increase	decrease	increase	increase	decrease
Speed or rate	increase	decrease	increase	increase	decrease
Oil Properties					
Viscosity	increase	increase	increase	decrease	increase
Surface tension	reduce	decrease	decrease	increase	decrease
Density	increase	increase	increase	decrease	increase
Gas Properties					
Density	increase	decrease	decrease	more even	decrease

Other important factors affecting the spray formation include injection pressure, combustion chamber air density, fuel properties (viscosity, surface tension, density) and spray orifice geometry. The effects of these factors on spray characteristics (7) are listed in Table 2.

Injection Nozzles

Injection nozzles have two functions: first, they atomize or finely divide the metered fuel to promote ignition and combustion; second, they distribute the fuel sprays through the combustion chambers with or without the assistance of air turbulence, to form a combustible mixture. In addition, the nozzles of a multi-cylinder engine should flow uniformly for equal fuel distribution to each engine cylinder. There are a wide variety of nozzle designs to provide the spray characteristics required for each type of combustion chamber, but all nozzles can be classified as "open" or "closed".

Open injector designs as illustrated in Fig. 8 are arranged so that injected fuel pressure acts on both sides of a check or poppet valve, and so these valves act as pressure reducing elements. The effective injection pressure is the pressure of the fuel entering the injector minus the valve closing pressure. Open injectors achieve only coarse atomization of fuel, which is satisfactory for pre-combustion chamber (indirect injection) engines. Their advantages are simplicity, absence of any fuel leak-off piping, and ease of servicing.

The basic feature of closed injectors is that fuel injection pressure acts on only one side of the valve. This is characteristic of a conventional closed type nozzle, shown in Fig. 9, in which only the lower end of the valve is explosed to the injection pressure while the upper end is vented via leak-off piping to the supply source. The valve is a lapped fit in its body so that only a slight amount of fuel, adequate for lubrication, leaks to the upper end of the valve.

Closed injector designs are used in direct injection combustion chambers because of their ability to generate finely atomized multiple fuel sprays, as required by this type of combustion chamber.

A complete assembly shown in Fig. 10 comprises the nozzle holder containing the spring and adjustment means for controlling the valve opening pressure, a spindle for transferring the spring load to the valve, fuel inlet and leak-off passages, nozzle valve and body, spray tip, and nozzle retaining nut. The valve lift is limited by the hardened stop plate interposed between the valve body and holder body.

In operation, metered fuel, under pressure from the injection pump, flows through the inlet, the edge filter, and around the valve, filling the nozzle body. When the pressure acting on the differential area (the difference between the area of the major diameter of the nozzle valve at the valve guide and the area at the valve seat) overcomes the spring force of the pressure adjusting spring, the nozzle valve lifts off its seat. As the valve raises to its predetermined lift height, high pressure fuel is allowed to flow through the spray orifices in the tip, details of which are shown in Fig. 11. When delivery to the nozzle ends and injection pressure drops below the preset nozzle opening pressure, the spring returns the valve to its seat.

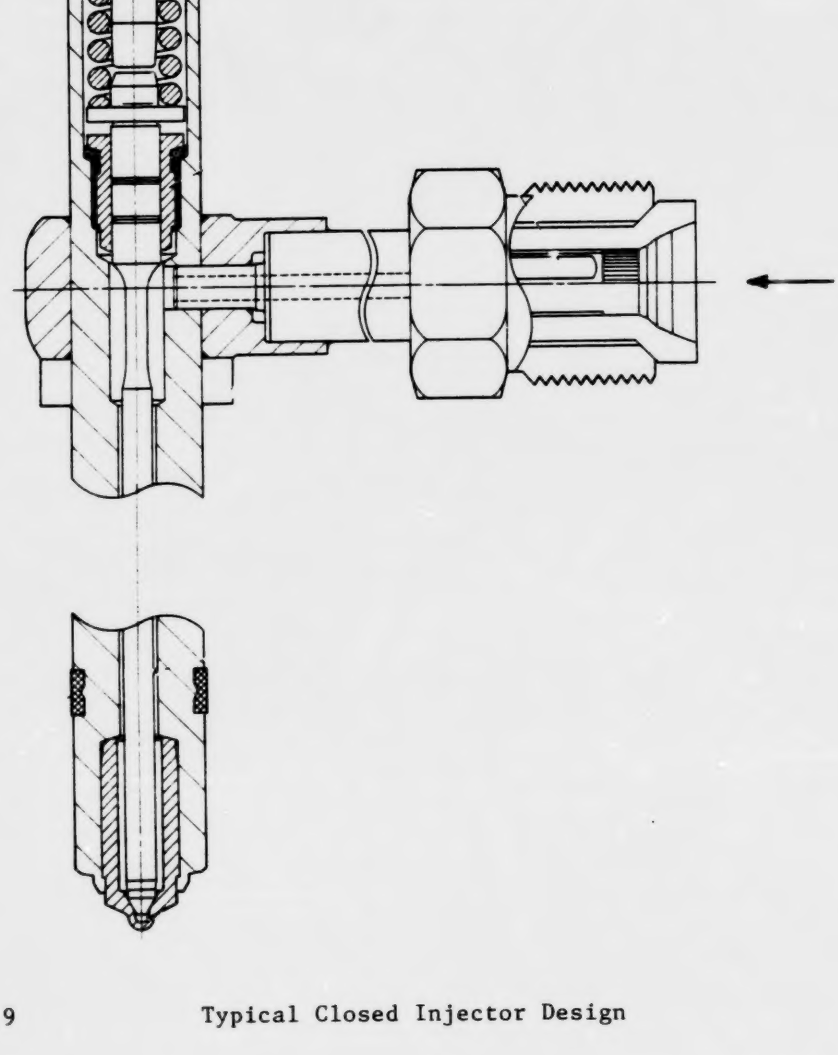

Fig. 9 Typical Closed Injector Design

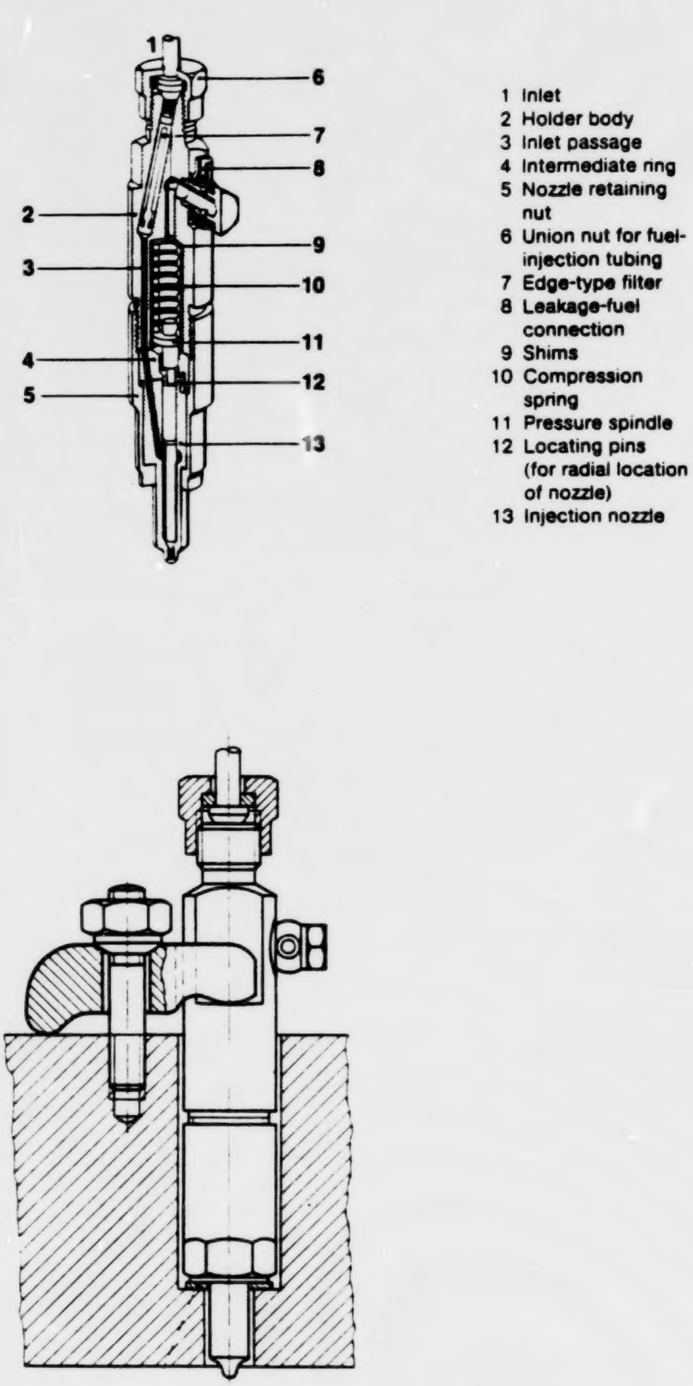
1
2
3
4
5
6
7
8
9
10
11
12
13
1 Inlet
2 Holder body
3 Inlet passage
4 Intermediate ring
5 Nozzle retaining nut
6 Union nut for fuel-injection tubing
7 Edge-type filter
8 Leakage-fuel connection
9 Shims
10 Compression spring
11 Pressure spindle
12 Locating pins (for radial location of nozzle)
13 Injection nozzle

Fig. 10 Alternative Closed Injector Design and Installation

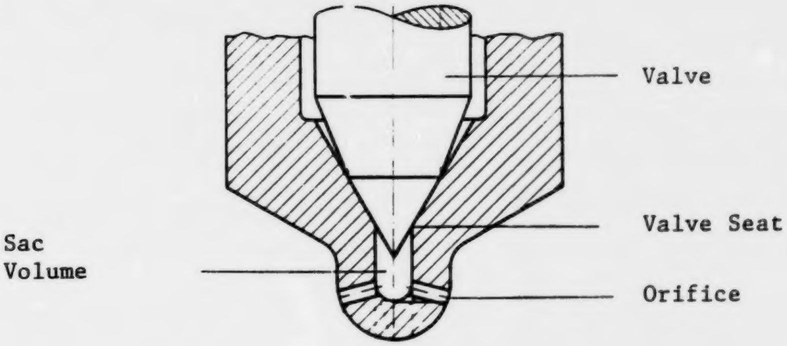
Valve
Valve Seat
Sac
Volume
Orifice

Fig. 11 Details of Spray Tip Construction

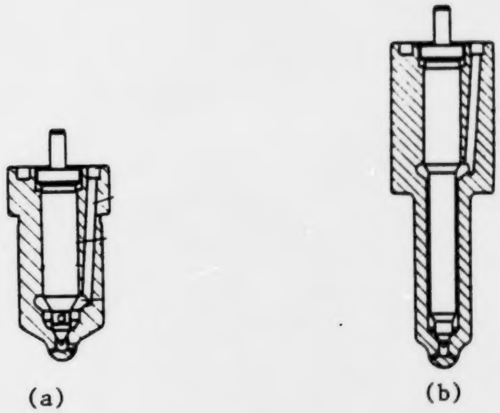

Fig. 12 Integral Spray Tip/Valve Body Designs

During injection, a small amount of fuel leaks through the clearance between the nozzle valve and its guide, lubricating and cooling all moving parts. The fuel flows through a leak-off boot at the top of the nozzle body and returns to the fuel tank.

In some closed valve designs as illustrated in Fig. 9 the valve is seated in an inserted tip, but a manufacturing difficulty then arises in obtaining the required close concentricities of the valve bore and seat. In Fig. 12a the spray tip is made an integral part of the valve body to overcome the alignment problem and to reduce the dead volume in the sac hole between the valve seat and orifice. This decreases dribbling from expansion of the fuel in the sac, but it brings the valve and its seat in closer proximity to the heat of combustion. Fig. 12b shows a similar nozzle design with the valve guide portion removed from the heat zone.

The nozzle opening and closing pressures should be sufficiently high to ensure that the valve seats against high combustion chamber pressures after the injection has ended; otherwise combustion gases will enter the nozzle fouling the orifices and valve with carbon.

Multiple orifice nozzles are most suitable for open chamber engines because of the versatility with which changes can be made to provide any desired penetration, dispersion, and duration of injection. The orifices can be drilled as small as 0.13 mm diameter or as large as 0.80 mm, and the number of orifices can vary from 4 to 10 depending on the engine bore and the amount of air swirl.

Nozzle Cooling

For satisfactory operation the nozzle should not be subjected to excessive temperatures during engine operation. For example, maximum operating temperatures near the nozzle tip are 150°C for standard case-hardened steel nozzles and 260°C for nozzles of heat resistant steel. Overheating is manifested by discoloration of the nozzle, erratic valve operation, troublesome carbonization of orifices and valves, and finally by stuck valves. Three methods have been used for nozzle cooling: direct, indirect, and internal cooling.

Direct nozzle cooling involves circulation of a liquid cooling medium (water or fuel) between the injector body and the bore in the cylinder head, with sealing accomplished by gaskets or O-rings. Direct cooling of the injector holder was the technique employed by UTME during propane testing on the laboratory engine, to eliminate heat transfer to the liquid propane during its passage through the injector.

Indirect cooling requires internal cooling passages in the injector holder and nozzle bodies. The coolant, usually fuel oil, enters through coolant connections in the holder and flows down and around the nozzle before returning through the outlet passage. This method is the most costly and complicated, and it is only used if the other methods are not satisfactory. These or directly cooled nozzles are usually recommended when residual fuel is used.

Control of nozzle temperatures in air cooled engines is more difficult than in the case of liquid cooled designs. This usually necessitates fins on the injector bodies and shrouds or flanged copper sleeves surrounding the nozzle extension to shield the end face of the nozzle from radiant heat and to conduct heat away from it.

Nozzle Coking

One of the troubles sometimes experienced under conditions of inadequate cooling or operation on heavy fuels is plugging of spray holes or build up of carbon formations on the nozzle tips, an occurrence termed coking.

This condition is apt to result if dribbling or post-injections occur, or if the nozzles have large sac volumes, so that the fuel trapped therein between injections undergoes decomposition from exposure to high temperatures. This decomposition of the fuel occurs at temperatures above 180°C and progresses faster as the temperature increases. It is influenced by the chemical composition of the fuel and the presence of catalytic materials, one of which is carbon. These formations are found mostly in slow speed, large bore engines burning residual fuels of 80 to 90 percent carbon and high ash content.

Coking of nozzle tips has also been observed when operating in a dual-fuel mode using two injectors. Coking of the pilot fuel nozzle occurs following prolonged operation at high load, when little or no injection of pilot fuel is taking place, resulting in decomposition of the stagnant fuel charge in the sac volume.

Plugging of nozzle orifices has been experienced when operating in a dual-fuel mode using two injectors.(8) During engine warmup on diesel fuel only, the primary fuel injector valve is forced open slightly under influence of cylinder pressure allowing small deposits of carbon to form in the sac volume.

Solutions to these problems are well known, and include such measures as the addition of cleansing additives to the fuel (9) and ensuring that at least a small amount of fuel is sprayed from both injectors at all times.

4.5 Design Requirements for Injection System

It was the subject of the present investigation project to research the diesel ignition of propane using a pilot injection of No. 2 diesel fuel to initiate combustion, employing a retrofit fuel injector capable of injecting two fuels through a standard engine cylinder head aperture.

The design requirements for such a fuel injection system as dictated by the fuel characteristics, engine type, engine application, and equipment retrofit capability, are summarized as follows:

Primary fuel	Propane Grade I (HD-5)
Pilot ignition fuel	ASTM No. 2 Diesel oil
Test engine type	Waukesha CFR 48 single cylinder
Compression ratio	Variable
Combustion chamber type	Pre-chamber
Engine displacement	610 cm^3
Engine speed, max.	1800 rpm
Primary fuel delivery, max.	38 mm^3 per injection
Pilot fuel delivery, max.	3.8 mm^3 per injection
Injection pressure, max.	21,000 kPa
Injection timing control	Fully variable, each fuel independent of other fuel timing

Injection system drive	Test engine driven. Provide auxiliary motor-driven variable speed drive for bench testing.
Pump lubrication	Required for propane pumping
Fuel supply pressure	Gravity head tank sufficient for No.2 diesel fuel pump; propane requires approx. 4000 kPa pressure at injection pump inlet to avoid cavitation
Fuel filtration	Both fuels require filters rated at 3-5 micron particle size retention ability
Fuel cooling	Water cooling jacket required on propane injector and propane delivery line
Dimensions of injector aperture in cylinder head	Firing face bore 14 mm diameter, stepped hole to external face bore 28.2 mm (see Fig. 13)
Instrumentation	Delivery line pressure transducer Injector valve displacement transducer
Injection system retrofit capability	Required

4.6 System Design and Fabrication

The system design constraints imposed by the particular requirements listed in the preceding Section 4.5 resulted in the construction of a prototype 2-fuel pumping and injection system based upon modified standard components. In addition to the obvious cost advantage of this approach as compared to the alternative design and fabrication of custom injection hardware, the use of stock items allowed some flexibility in terms of ad hoc component substitution during final system development testing. The ability to quickly replace damaged or failed components was also beneficial from a schedule point of view.

The requirement for separate propane and diesel oil injection at high pressure (max. approx. 21,000 kPa) with completely independent timing and volumetric delivery adjustments was satisfied by mounting two single-plunger injection pumps inline, connected by a variable phase angle mechanical coupling. The drive was provided by a takeoff shaft coupled to the test engine crankshaft through another variable phase angle mechanical coupling.

Propane was maintained in the liquid state by controlling its temperature and pressure within the critical values listed in Table 1 (refer to "Vapor Pressure"). Cavitation problems were anticipated at the pump plunger inlet (suction stroke) and in the delivery line adjacent to the engine. The design solution chosen was to circulate cooling water through a jacket surrounding the propane delivery line and the fuel injector itself. A pressure level of approx. 4000 kPa was maintained at the propane pump inlet using a piston-type priming pump fitted with a pressure relief valve to permit recirculation of excess propane.

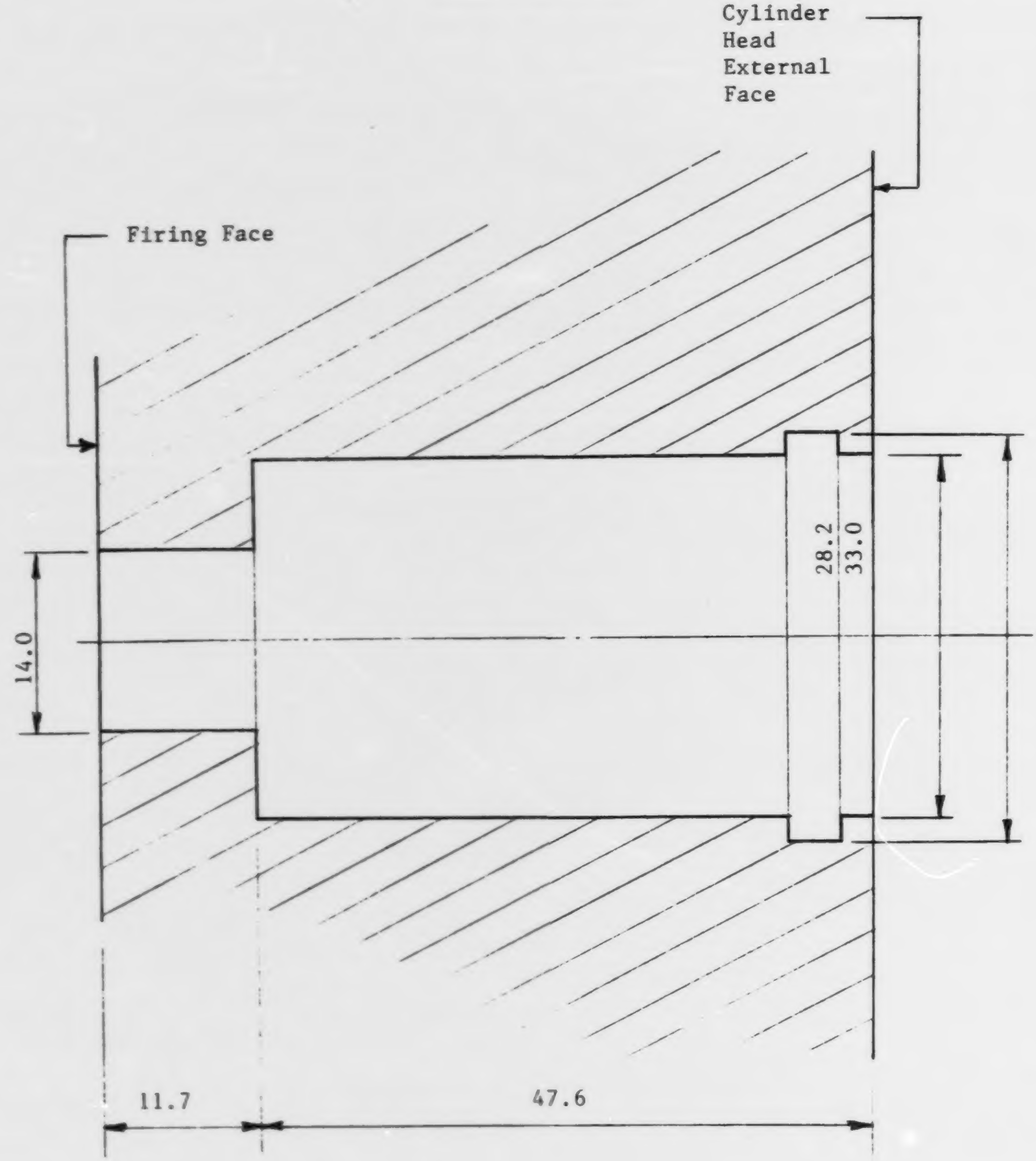
Cylinder Head External Face
Firing Face
14.0
28.2
33.0
11.7
47.6

Fig. 13 Injector Aperture in Waukesha CFR 48 Cylinder Head

The issue of propane's poor lubricity was addressed at the time of system design. It was postulated that potential injection pump plunger wear would not be excessive if gas film lubrication conditions could be achieved. In practice, catastrophic wear/failure of both priming pump and injection pump plungers was experienced. Effective design solutions were found by converting the priming pump to diaphragm operation and installing an externally-pressurized oil lubrication system on the injection pump. Both these measures are accepted practice on gasoline high pressure injection systems, since gasoline is also a very poor lubricant.

When selecting a 2-fuel injector design, a serious challenge was presented by the combination of constraints imposed by the test engine injector aperture size (refer to Fig. 13), required injection pressures, and the requirement for two separate fuel sprays (the latter deemed necessary to permit full experimental variability of spray patterns, quantity and timing).

Numerous designs employing single poppet valves and single needle-type valves were prepared, with provision for dual-fuel delivery via two fuel-feed drillings in the injector body leading to a common nozzle. These designs were not pursued beyond the conceptual stage, since the requirement for two separate fuel sprays was considered a priority in this development program.

The injector design eventually adopted was based on the use of two separate spray nozzles installed in a common injector body, with injection initiation and cutoff regulated by individual slender needle valves for each fuel stream. In order to avoid interference between valve assemblies in the valve guide and fuel inlet areas, the valve longitudinal axes were inclined at a small included angle. Stock nozzle spray tips were employed, machined flat on one side to permit closer spacing of the valve centerlines. Production pencil injector components (Stanadyne) were used for fuel inlet, valve, valve guide and spring assemblies. An example of a preliminary injector designed and fabricated following this approach is illustrated in Appendix C.

Maintaining adequate body wall thickess in the vicinity of the spray nozzle proved to be a difficult problem, given the requirement to limit the injector body minor diameter to 14 mm. Wall failure (bursting) was experienced on early versions of the injector, even at moderate injection pressures.

Another problem encountered was erratic seating of the valve on its seat, apparently due to misalignment of the axes of the valve guide and seat, in spite of careful machining. Injector manufacturers confirmed the requirement for dedicated machining equipment for this task.

To circumvent the known fabrication obstacles of maintaining adequate body wall thickness and correct alignment of the valve assembly, the bodies of stock pencil injectors (Stanadyne) were retained, with reduced outside diameter, and pressed into a common holder using a temperature-resistant epoxy adhesive. A heat curing cycle then followed to properly age the epoxy adhesive. Details of the final configuration are shown in Appendix C.

A survey of available pencil nozzle spray hole sizes resulted in the selection of nozzles having 2 holes of 0.28 mm dia. for the propane injector, and 1 hole of 0.18 mm dia. for the diesel pilot injector. These values were selected based upon the fuel properties listed in Table 1 and delivery requirements stated in Section 4.5.

Several injectors were fabricated for engine testing, each with a different spray orientation. The different spray patterns were achieved by selectively plugging unwanted spray holes in the stock nozzles, using an adhesive and tapered wire of the appropriate diameter.

Fabrication and bench testing of injectors was performed by IVI in conjunction with specialist suppliers. The 2-fuel pumping system was designed and built by SWRI, and then shipped to IVI for the purpose of injector development and bench testing, using a variable speed electric motor drive. The complete system was then installed on the laboratory test engine at UTME, where a program of engine performance testing was conducted.

5.0 2-FUEL INJECTOR ENGINE TESTING

5.0 DUAL-FUEL INJECTOR ENGINE TESTING

5.1 Engine Testing Methodology

The engine test program executed by UTME using a single-cylinder laboratory engine was organized to accomplish the following objectives:

- determine the performance characteristics of the test engine operating in two dual-fuel modes (fumigated, and injected propane with pilot diesel ignition) compared to performance levels using No. 2 diesel oil only

- define guidelines for equipment requirements for a full-scale injection system capable of handling liquid propane and No. 2 diesel oil, operating on the pilot ignition principle.

Initial tests to establish baseline performance levels of the laboratory engine were performed using No. 2 diesel fuel fed to the engine through the standard injector (poppet-nozzle type) and pumping system. Baseline tests were carried out on a dynamometer at three compression ratios (19:1, 22:1 and 25:1) and at three engine speeds (1200, 1500, and 1800 RPM). From the measured data, calculations were made of engine torque, fuel consumption, exhaust smoke opacity, carbon monoxide, hydrocarbons, and oxides of nitrogen emissions. All tests were conducted with the engine inlet unsupercharged, and at inlet air temperatures not exeeding 42°C.

The dynamometer tests were repeated covering the same range of test points with the engine operating in two dual-fuel modes, ignition being provided by a pilot injection of No. 2 diesel oil in both cases. The dual-fuel tests were conducted using two means of propane (primary) fuel introduction; the first being fumigation, in which gaseous propane at ambient pressure was supplied to the test engine's inlet manifold where it was mixed with air entering the cylinder, and the second means being liquid injection, in which liquid propane was pumped to a high pressure and injected directly into the engine's prechamber concurrent with the pilot injection of diesel oil.

The latter testing involving liquid propane injection was performed using two separate injectors for propane and diesel oil followed by a series of tests utilizing the 2-fuel injector supplied by SNC/IVI described in Section 4.6. Additional parameters varied during this series of tests included injection timing (absolute and relative), injector valve opening pressure, and injector spray orientation.

Following completion of engine testing, the results were analyzed and performance calculations made for all modes of fuel introduction. Typical fuel consumption data using the 2-fuel liquid propane injector were used to calculate the projected fuel cost savings when applying this dual-fuel performance to the full-scale case of a heavy road vehicle. The results of this calculation, tabulated for various propane and diesel fuel prices and different assumed annual levels of consumption, are presented in Section 6.0.

5.2 Test Equipment Description

Major items of equipment used to execute the UTME engine test program included the following:

- single-cylinder variable compression ratio Waukesha CFR 48 engine having 610 cm^3 displacement, equipped with a pre-chamber cylinder head and installed on an electric power absorption dynamometer
- dual-fuel injection pump assembly for propane and diesel oil driven by the engine crankshaft, with independent controls for each fuel delivery and injection timing. Auxiliary pumping systems were used to generate priming pressure on the propane circuit and to provide lubricating oil to the plunger of the propane injection pump. The propane circuit downstream from the pump inlet was maintained above 4000 kPa and was modified as required to avoid any source of propane leaks.
- fuel consumption measuring apparatus operating on a storage tank weight-loss method
- injection line pressure and cylinder pressure transducers
- injector valve lift transducers
- crankshaft angle transducer
- emission measuring equipment

A detailed description of all equipment and instrumentation used during testing is included in the UTME test report contained in Appendix D.

5.3 Test Results

The engine tests successfully demonstrated that liquid propane could indeed be injected into a diesel engine and ignited using a pilot injection of diesel fuel, and that the energy conversion performance is comparable to that obtained with No. 2 diesel fuel.

Both propane liquid injection and fumigation techniques resulted in engine operation with significantly reduced solid particulate (smoke) emissions and moderate reductions in NO_x emissions, as compared to No. 2 diesel baseline data.

The reduction in exhaust smoke was particularly apparent at higher power levels, indicating that a propane/diesel engine's output could be increased without exceeding smoke limits established for operation on diesel fuel.

Engine operation using propane liquid injection required an average of 10% pilot diesel fuel (total fuel energy basis) for complete and reliable combustion, while operation using fumigated propane required average diesel fuel levels of 15 - 25%.

Hence operation at present fuel price levels would be more economical using propane liquid injection compared to fumigation.

It was further noted that engine operation using liquid propane injection was more stable than when using fumigation, with little tendency to surge or misfire.

Similar fuel consumption figures on a total energy basis were recorded for both dual-fuel techniques and No. 2 diesel operation, with consumption ranging from 7.5 to 9.0 MJ/kWh.

Complete engine test results and analyses, prepared by UTME, are presented in Appendix D.

6.0 PRELIMINARY MARKET ASSESSMENT

6.0 PRELIMINARY MARKET ASSESSMENT

6.1 Application Range

The propane/diesel fuel injection system tested under the present program is intended to be developed into a retrofit or original equipment system applicable to diesel trucks operating on freight and other long-haul routes throughout North America where propane is readily available.

Market penetration is foreseen in cases where the following criteria are satisfied:

- sufficient onboard vehicle space exists to accommodate the addition of propane fuel storage tanks. On many long-haul trucks, extra diesel fuel tank capacity has been added to take advantage of regional differences in the price of diesel oil. Removal of these extra tanks would permit the installation of adequate propane storage volume. Straight trucks present less challenge regarding tank installation, since most designs have available space alongside both longitudinal chassis frame rails. Straight truck tank installation is further facilitated since the required propane storage volume is less than for long-haul trucks, due to the shorter routes travelled by most straight trucks.

- the engine is equipped with an externally-mounted fuel injection pump, as installed on diesel truck engines manufactured by Carterpillar, Daimler-Benz, Deutz, Ford, Hino, International, Isuzu, Iveco, Komatsu, Lister, MAN, MWM, Mercedes-Benz, Mack, Nissan, Perkins, Renault, Rolls-Royce, and Volvo-Scania.

- sufficient annual fuel consumption or user demand is present to generate the required cost savings from 2-fuel operation to permit a reasonable payback period for the additional hardware costs involved

- sufficient differential exists between the price of propane and No. 2 diesel fuel

6.2 Fuel Injection System Components

A fully developed propane/diesel fuel injection system suitable for retrofit or original equipment installation on large diesel trucks is illustrated in block diagram format in Fig. 14 and would involve the addition of the following principal components:

Propane fuel tanks
Propane priming pump
Propane transfer piping to pump
Propane fuel injection pump with drive
Lubricating oil supply
Electronic controller for pumps
Engine-mounted sensors
Throttle linkage
Diesel and propane injection tubing
Dual-fuel injectors
Miscellaneous brackets and fittings

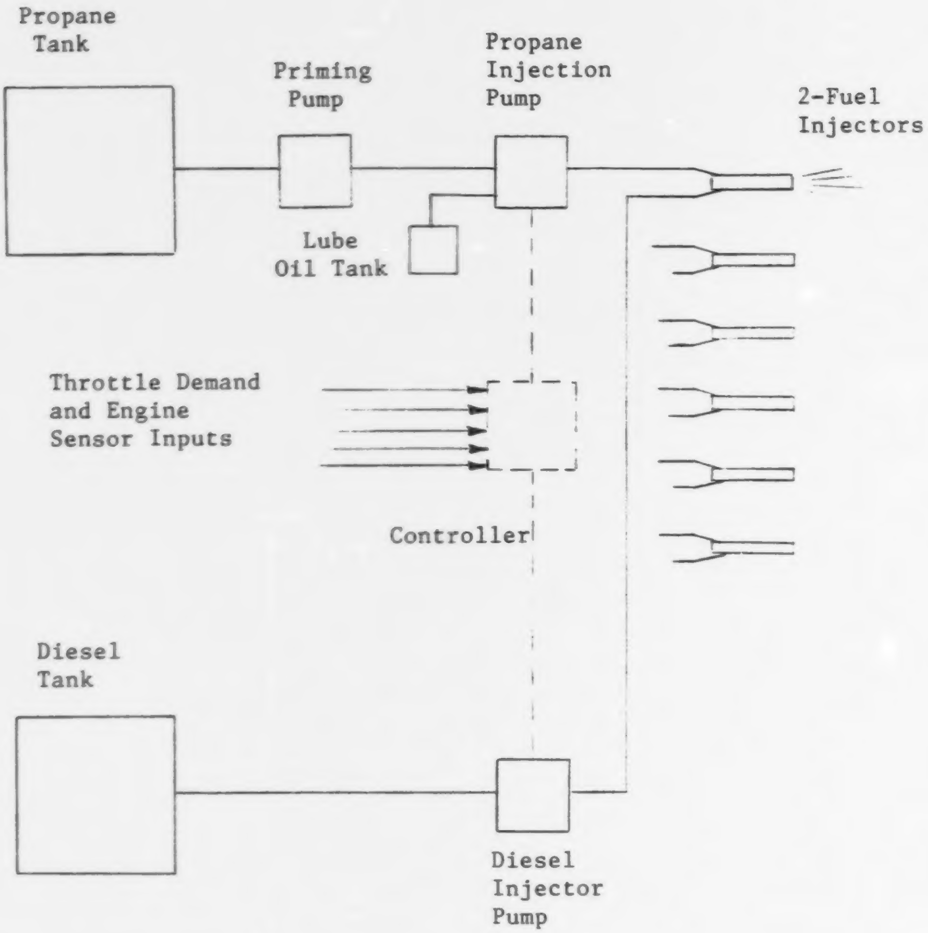
Propane Tank
Priming Pump
Propane Injection Pump
2-Fuel Injectors
Lube Oil Tank
Throttle Demand and Engine Sensor Inputs
Controller
Diesel Tank
Diesel Injector Pump

Fig. 14 Schematic of Vehicle Propane/Diesel Fuel Injection System

6.3 Fuel Injector Manufacturing Cost Evaluation

A manufacturing plan for the production of various annual quantities of 2-fuel injectors using dedicated tooling and machinery has been prepared by IVI Inc. and is presented in Appendix E. The evaluation, which includes a description of the required fabrication steps, fabrication tooling, assembly tooling, time and cost, is applicable to production runs of up to 100,000 injectors (corresponding to the conversion requirements of approximately 15,000 engines).

The approach taken to facilitate the cost estimation was to assume that the 2-fuel injector would incorporate certain key components derived from standard injectors. In this way, the maximum expected manufacturing costs could be solidly established, i.e. the approach followed conservative estimating practice.

Several of the fabrication, assembly and testing operations require the purchase and/or adaptation of precision, dedicated machines. Existing equipment and skills are available in the market place to accomplish all of the foreseen tasks given sufficient development engineering and staff training inputs.

Unit costs for 2-fuel injectors were established at $700 for quantities up to 1000, based upon quotations from a machine toolmaker having the required skills to produce functioning prototypes. Costs for higher production levels, estimated by IVI, were $470 and $360 per unit for volumes of 10,000 and 100,000 units respectively.

6.4 Economic Analysis

The economic criteria listed in Section 6.1 together with the engine performance test results were applied to evaluation cases representing different regional fuel price situations across Canada. Key parameters in this evaluation include the following:

- average test engine fuel consumption operating on No. 2 diesel fuel. From the body of test data produced, an ISFC (indicated specific fuel consumption) value of 7.5 MJ/kWH was selected as a representative average of baseline performance.

- average test engine fuel consumption operating in the liquid propane injection 2-fuel mode. As some scatter was present in the resulting data, two values of ISFC were utilized in the evaluation, representing low and high average fuel consumptions (7.5 and 8.0 MJ/kWh respectively).

- diesel injection rate (the proportion of pilot diesel fuel required for stable combustion) which averaged 10% diesel, 90% propane, on an energy basis.

- lower heating values (LHV) of No. 2 diesel and propane (44.2 and 46.3 kJ/kg respectively).

- specific gravities of No. 2 diesel and propane (0.85 and 0.50 respectively) at 20°C.

- annual energy consumption, assumed for two different cases representing long-haul trucks (100,000 L per year) and also suburban trucks (50,000 L per year).

- regional retail pump prices at major centres across Canada for No. 2 diesel and propane fuel.

The baseline fuel efficiency of the test engine was assumed to be similar to the fuel performance of a full-sized multicylinder truck engine. The relative fuel efficiency figures when operating in both diesel and 2-fuel modes were then extended to compute annual fuel costs.

The results of such calculations are presented in Tables 3 and 4 for 8 Canadian regions for which propane and diesel fuel service station pump prices are published. The tabulated results indicate $1800 to $2900 annual fuel savings using the 2-fuel approach for operators of suburban trucks in and around Toronto, Edmonton and Vancouver. In the case of operators of trucks on long-haul routes passing through Toronto, Edmonton and Vancouver, fuel bills would be further reduced using 2-fuel equipment, with annual fuel savings ranging from $3700 to $5800.

A typical propane/diesel injection system conversion kit is estimated (10) to cost in the range of $6000 installed, employing stock and adapted standard components as described in Section 6.2.

The economic benefits accrued from 2-fuel operation would be sufficiently large to return the original cost of investment in fuel conversion equipment within a period ranging from one to three years, depending upon the route structure and location. This finding is supported by the limited acceptance of fumigated 2-fuel (propane/diesel) systems in some areas of Canada and the United States, despite their lower diesel displacement capability compared to the injected propane system.

TABLE 3 SUBURBAN ROUTE FUEL COST COMPARISON

Fuelling Location	Fuel Price ($/L)		Average Engine Fuel Consumption (MJ/kWh)	Annual Fuel Cost ($) for 50,000 L diesel energy equiv.	
	P	D		2-Fuel *	D
Halifax	0.423	0.554	Low	33 604	27 200
			High	35 844	-
St. John	0.375	0.552	Low	30 095	27 600
			High	32 101	-
Montréal	0.396	0.504	Low	31 386	25 200
			High	33 478	-
Toronto	0.271	0.480	Low	22 154	24 000
			High	24 151	-
Winnipeg	0.297	0.448	Low	23 888	22 400
			High	25 482	-
Regina	0.259	0.427	Low	21 014	21 350
			High	22 415	-
Edmonton	0.196	0.382	Low	16 196	19 100
			High	17 276	-
Vancouver	0.257	0.471	Low	21 088	23 550
			High	22 494	-

Note

1. P = propane D = No. 2 Diesel
2. Low = 7.5 MJ/kWh High = 8.0 MJ/kWh
3. Fuel prices shown are 1985 pump averages, courtesy of EMR, Commodities Division
4. * based on 10% diesel, 90% propane (energy basis)

TABLE 4 LONG-HAUL ROUTE FUEL COST COMPARISON

Fuelling Location	Fuel Price ($/L)		Average Engine Fuel Consumption (MJ/kWh)	Annual Fuel Cost ($) for 100,000 L diesel energy equiv.	
	P	D		2-Fuel *	D
Halifax	0.423	0.554	Low	67 207	54 400
			High	71 688	-
St. John	0.375	0.552	Low	60 189	55 200
			High	64 202	-
Montréal	0.396	0.504	Low	62 772	50 400
			High	66 956	-
Toronto	0.271	0.480	Low	44 307	48 000
			High	49 102	-
Winnipeg	0.297	0.448	Low	47 777	44 800
			High	50 963	-
Regina	0.259	0.427	Low	42 028	42 700
			High	44 830	-
Edmonton	0.196	0.382	Low	32 393	38 200
			High	34 552	-
Vancouver	0.257	0.471	Low	42 176	47 100
			High	44 987	-

Note

1. P = propane D = No. 2 Diesel
2. Low = 7.5 MJ/kWh High = 8.0 MJ/kWh
3. Fuel prices shown are 1985 pump averages, courtesy of EMR, Commodities Division
4. * based on 10% diesel, 90% propane (energy basis)

New technological developments (11, 12) have taken place in the field of fuel injectors controlled directly by electronic means, making obsolete the conventional governor-controlled fuel injection pump. With these designs, recently introduced to the market, the injector is responsible for fuel metering and injection timing tasks, with injection pressure being provided by a low cost hydraulic pump.

An electronic system has great potential for cost reduction if applied to a suitably designed 2-fuel injector, since the most expensive element, the injection pump with governor, would not require duplication. All of the known attributes of electronic control would be available (ease of programming and reprogramming, low tooling cost, compact dimensions) with the additional benefit of convenient switchover to and from dual-fuel operation.

7.0 DISCUSSION

7.0 DISCUSSION

The engine tests have shown that liquid propane can be injected directly into a diesel engine cylinder through injectors of conventional design, and that the energy conversion performance is close to that of a diesel fuel system. The indications are that with optimization of the prototype system tested engine operation would be as efficient as that achieved with developed diesel fuel systems.

Both gas-fumigated and liquid injected propane test results show large reductions in solid particulate emissions as compared to diesel fuel operation, as would be expected. This will permit the use of smaller engines operating at higher specific outputs than at present, and has special implications regarding diesel engines in urban service.

Test measurements were all made under steady state conditions and as such it was possible to achieve large (75 - 85%) displacement of diesel fuel with the fumigation system. The limitations to accurate metering and reliable ignition using fumigation during low load transient conditions are well known and have not been resolved. There is every reason to believe that by injecting liquid propane the necessary control could be exercised as is the case with diesel fuel injection systems.

Tests were all conducted on a Waukesha CFR prechamber engine. An open-chamber engine would certainly pose more stringent requirements on a twin injector design but the extent can only be assessed by further testing.

The diesel engine development work conducted reconfirmed the more commonly experienced problems with propane handling at lesser pressures: all connections, seals and precision-mated surfaces are particularly prone to leakage at diesel system pressures.

A closed system is therefore virtually mandatory. Due to the low lubricity of propane some form of injection pump lubrication system is necessary, and injector wear also had to be resolved during development. It was noted that solutions to the problems encountered were all well known.

Propane has also been identified with exhaust valve and seat damage (recession). The performance tests undertaken did not involve endurance testing, but it would appear possible that controlled injection could in turn be used to ease problems due to excessive pressure rise rates and undesirable temperature distributions. Whether the presence of the pilot diesel fuel would also assist in reducing valve recession is not known.

The diesel fuel could, however, be used in any developed 2-fuel system to assist with propane cooling if necessary. This should not pose any particular problems since the technology is available from gasoline injection systems.

The result of the design, development and test experience has been to show that propane can be injected as a liquid into a diesel engine which will then operate as efficiently as would be the case on diesel fuel alone. In geographic areas such as Toronto, Edmonton and Vancouver the economics of a developed system would be of interest to fleet operators of suburban and long-haul trucks.

The principles involved in dual fuelling propane/No. 2 diesel are directly applicable to other fuel combinations such as off-spec. diesel/No. 2 diesel or methanol/No. 2 diesel.

A general interest in the concept is evidenced by the design activities of other manufacturers in this area revealed by the technology studies.

8.0 CONCLUSIONS AND RECOMMENDATIONS

8.0 CONCLUSIONS AND RECOMMENDATIONS

- The viability of dual-fuel propane injection/No. 2 diesel engine operation has been demonstrated.
- Liquid injection propane/No. 2 diesel performance compares favourably with baseline diesel performance.
- Solid particulate emissions are radically reduced compared to operation on No. 2 diesel fuel.
- Testing and development should be extended to open-chamber engines, and should include durability testing.

APPENDIX A

REFERENCES

REFERENCES

(1) G.A. Karim, "A review of Combustion Processes in the Dual Fuel Engine - the Gas Diesel Engine", Prog. Energy Combustion Sci. Vol. 6, March 1980.

(2) E. Ecklund et al., "State of the Art Report on the Use of Alcohols in Diesel Engines", SAE paper 840118.

(3) "Enterprise R4 Engine Runs on Coal-Derived Fuel", Modern Power Systems, Feb. 1983.

(4) K. Bro, P.S. Pedersen, "Alternative Diesel Engine Fuels: An Experimental Investigation of Methanol, Ethanol, Methane and Ammonia in a D.I. Diesel Engine with Pilot Injection", SAE paper 770794.

(5) F. Pischinger, R. Menne, "LPG-Diesel Dual-Fuel Operation with Liquid Injection of Propane/Butane into the Intake Manifold", MTZ Motortechnische Zeitschrift 40 (3), 1979.

(6) T. Seko et al., "Methanol Diesel Engine and Its Application to a Vehicle", SAE paper 840116.

(7) P.G. Burman, F. DeLuca, Fuel Injection and Controls for Internal Combustion Engines, Simmons-Boardman Publishing Corp., 1962.

(8) J.T. Gray, et al., "Ammonia Fuel - Engine Compatibility and Combustion", SAE paper 660156.

(9) R.E. Olsen et al.," A Fuel Additive Concentrate for Removal of Injector Deposits in Light-Duty Diesels", SAE paper 841349.

(10) R.F. Webb Corp. Ltd., "An Investigation of Propane as a Motor Fuel in Canada", Feb. 1982, p. 114.

(11) "Avco Lycoming, Deere Will Develop Stratified-Charge Rotary Engine", Aviation Week and Space Technology, 1 April 1985.

(12) G. Kattenbuch et al., "Design and Operating Method of Electronic Injection System, " MAN Technical brochure.

APPENDIX B

ADVANCED INJECTION SYSTEMS PATENT DESCRIPTIONS

- Spill valve for I.C. fuel injection system - has fast response time and controls injection and metering of fuel through wide engine speed range.

  Registry: US

  Comment: May not apply to compression ignition engines.

- Reciprocating plunger type fuel injection pump - has balanced electromagnetically operated control valve requiring low operating forces.

  Registry: CA, GB

  Comment: An important control element. CA registry unusual.

- Fuel injector system for compression ignition engines - uses fuel pressure to open and close injector valves.

  Registry: DE, FR, US, GB, CA, US, CH, IT, DE

  Comment: Important design element. Similar to Lucas patent.

- Fuel injection jerk pump for I.C. diesel engines - utilizes electronically controlled valving to control timing of injection and metering of fuel quantity for each injection.

  Registry: EP, US, CA

  Comment: Uses accumulator and is similar to proposed SNC system.

- Single solenoid floating piston distributor pump - controls both fuel quantity and injection timing by single solenoid operated control valve.

  Registry: EP, BR, CA

- Fuel injector distribution pump for I.C. engine - has single solenoid valve in pump spill path to control timing and duration of injection pulses.

  Registry: EP, BR, US, CA, EP, DE

  Comment: Spill timing control similar to proposed SNC system.

- I.C. engine fuel injection system - has first and second rotating valves controlling stopping and starting of pilot and main injectors with different fuels.

  Registry: EP, WO, US, JP

  Comment: 2-fuel injection system.

- Springless fuel injection nozzle - has differential fuel supply pressure metered for operation of valve without resonance.

  Registry: US, WO

  Comment: Similar to Bendix and Lucas patents.

- Fuel injection appts. for I.C. engine - has rotary valves for starting and stopping both pilot and main injection through two nozzles.

  Registry: WO, EP, US, JP

- Fuel injection appts. for I.C. engine - provides for pilot injection using two electrically-driven rotating valves with single injector and pump.

  Registry: EP, WO, JP, CA

  Comment: Permits staged injection of single fuel.

- Rotary valve fuel injector for I.C. engine - has twin shouldered rotary valves to intermittently block supply conduit for reduced vibration.

  Registry: WO, EP, JP, CA, US

  Comment: Permits single pulse or staged injection of single fuel using constant pressure supply pump.

- Accumulator fuel injection nozzle - has inwardly opening needle-type outlet valve responsive to pressure in chamber with dump valve.

  Registry: US

- Stratified charge rotary I.C. engine - has pilot injector whose ignited fuel ignites that of main injector.

  Registry: US, CA

  Comment: technology purchased by Deere Co.

- Diesel engine with dual fuel injection system - has waste secondary chlorinated hydrocarbon fuel injected marginally before diesel flow.

  Registry: US

  Comment: System intended for disposal of waste hydrocarbons having poor compression ignition characteristics.

- Fuel injection nozzle for dual fuel engine - has pressure unloading valve for main fuel which incorporates unloading volume.

  Registry: EP, DE, US, EP, DE

  Comment: Permits injection of two fuels through a common injector.

- Fuel injection nozzle for I.C. engine - has second fuel passage for pilot fuel injection.

  Registry: DE

- Liquid fuel injection equipment - has auxiliary passage in distributor delivering before main discharge passage with outlet valve.

  Registry: DE, GB, FR, US, GB

  Comment: Could permit staged injection of single fuel.

- I.C. engine fuel injection system - has two piston and cylinders both with non-return valves, movement of second piston imparting movement to first.

  Registry: GB, FR, US, GB

- Fuel injection system for diesel engine - has valve operating piston admitting initial quantity before main fuel dose.

  Registry: DE, FR, US, GB, SU

  Comment: Permits staged injection of single fuel.

- Fuel injection system for diesel engine - has independent injection systems to meter alternative fuels for optimum performance.

  Registry: DE, FR, US, GB

  Comment: 2-fuel injection system.

- Fuel injection system for pressure ignited engine - uses two-stage fuel injection controlled by valve unit with three control position.

  Registry: DE, FR, US, GB, SU

- Pilot fuel injector system for rotary distributor pumps - has pump rotor with auxiliary plungers in front of main plungers.

  Registry: DE, FR, US, GB

  Comment: Permits staged injection of single fuel

- Fuel injection system injector valve - is opened by high hydraulic pressure and closed by low hydraulic pressure.

  Registry: DE, FR, US, GB, IT

  Comment: Hydraulically-actuated injector allows considerable design flexibility.

- Fuel injector system for compression ignition engines - has injector valve opened and closed by fuel pressure differences.

  Registry: DE, FR, BR, ZA, US' CA, GB, IT

  Comment: Similar to #7. Note extensive patent registry list. Similar to Bendix patent.

- Pressure operated fuel injector nozzle - has two injector needle valves to give stratified charge.

  Registry: DE, FR, IT

- Fuel system for multicylinder I.C. engine - has two injector inlets for pumped fuel, main injection and small quantity pilot injection phased in advance.

  Registry: GB, DE, FR, SU, US, GB

- Compression ignition I.C. engine - has pilot injection system for distributor pump with electrically operated auxiliary plunger.

  Registry: DE, FR, US, GB

- Numerous accumulator-based injectors with novel means of control have been patented.

- Single-plunger fuel injection pump - for I.C. engines providing successive injection stages.

  Registry: GB

  Comment: Could permit multiple injection pulses.

- Ethanol fuelled diesel engine - has diesel oil injector and distributor actuated by high ethanol pressure to inject oil for ignition at low pressure.

  Registry: DE, BR

  Comment: Full description probably in German. Brazil is market focus.

- Multi-fuel diesel I.C. engine - injects main fuel from perimeter and ignition fuel commencing later and from centre of combustion chamber.

  Registry: DE, BR

- Injection system for dual fuel diesel engine - has main injector with adjustable pressure divider for pilot injection.

  Registry: DE, BR

- Dual fuel diesel engine - has high pressure alcohol pump with indirect connection to increase pressure of diesel fuel.

  Registry: DE

  Comment: Fuel system circuit very similar to proposed SNC system.

- Diesel injector fuel injector - has two needle valves with stops adjustable via rotary face cam, for cold start and hot running.

  Registry: GB, DE

- Diesel engine fuel injection system - has pump and jet for each cylinder with pump pistons driven by servo pistons.

  Registry: CH, US, GB

  Comment: Concept similar to proposed SNC system.

- Fuel injector with two needle valves - has nozzle valve larger than control valve, both are initially moved together against spring pressure.

  Registry: DE

- Engine injection equipment with pilot pump - has adjustable hydraulic stop for pilot plunger return stroke.

  Registry: DE

  Comment: Permits staged injection of single fuel.

- Engine fuel injection system - has adjustable top and bottom stops for pilot injection pump plunger.

  Registry: DE

  Comment: Permits staged injection with one injection pump.

- Fuel system with pilot injection - has auxiliary pump plunger forming part of valve controlling main injector fuel.

  Registry: DE, GB

- I.C. engine fuel injection system - has pilot injection plunger driven by coaxial intermediate piston between union and nozzle.

  Registry: DE, GB, US

  Comment: Compact dimensions permit incorporation within space envelope of conventional injector.

- Engine fuel injection system - has accumulator plunger determining time interval between pilot and main injections.

  Registry: DE, EP, US

  Comment: Flexible delivery system similar to proposed SNC concept.

- Direct fuel injection for diesel I.C. engine - injects diesel fuel, then fuel-alcohol mixture, then again diesel fuel, in 3-phase sequence.

  Registry: DE, GB, FR, US

  Comment: Fuel pumping system not described.

- Adjustable fuel injector for dual fuel engine - has separate rotors to control fuel pressurization, with separate control sleeves regulating injection quantity of each fuel.

  Registry: WO

  Comment: A 2-fuel injection system. Goloff has assigned other patents to Caterpillar.

- Dual fuel engine, injection pump - has fuels metered into top and bottom pump chambers separated by plunger.

  Registry: DE

  Comment: Permits high pressure injection of 2 fuels into combustion chamber using a single pump.

- Arrangement for converting diesel engine to run on alternative fuel - comprises dual ducted injector substituted for conventional injector, and added-on constant delivery injection pump.

  Registry: ZA

- Diesel injector with separate nozzle - permitting minimal fuel residue after injection.

  Registry: SU

- Compression ignition I.C. engine - has two fuel injectors arranged for running on mixture of high and low grade fuels.

  Registry: DE, GB, BR, FR, ZA, US, GB, DE, US

- Staged direct injection diesel I.C. engine - includes main and pilot fuel injectors, with pilot angled toward piston centreline.

  Registry: US

  Comment: Developed from research funded by AAR.

- Accumulator-type diesel injection nozzle - has additional spring-loaded valve located in channel running to accumulator chamber for sharp fuel cutoff.

  Registry: SU

snc

APPENDIX B - COUNTRY CODES

These codes follow the ISO standard 3116-1974 (E)

B.1 COUNTRY CODE ORDER

AD	ANDORRA
AE	UNITED ARAB EMIRATES
AF	AFGHANISTAN
AG	ANTIGUA
AI	AFARS AND ISSAS
AL	ALBANIA
AN	NETHERLAND ANTILLES
AO	ANGOLA
AQ	ANTARCTIC
AR	ARGENTINA
AS	AMERICAN SAMOA
AU	AUSTRALIA
BB	BARBADOS
BD	BANGLADESH
BE	BELGIUM
BG	BULGARIA
BH	BAHRAIN
BI	BURUNDI
BO	BOLIVIA
BR	BRAZIL
BS	BAHAMAS
BT	BHUTAN
BU	BURMA
BV	BOUVETOYA
BW	BOTSWANA
BZ	BELIZE
CA	CANADA
CC	COCOS ISLANDS
CF	CENTRAL AFRICAN REPUBLIC
CG	CONGO
CH	SWITZERLAND
CI	IVORY COAST
CK	COOK ISLAND
CL	CHILE
CM	CAMEROON
CN	CHINA
CO	COLOMBIA
CR	COSTA RICA
CS	CZECHOSLOVAKIA
CT	CANTON AND ENDERBURG ISLANDS
CU	CUBA
CV	CAPE VERDE
CX	CHRISTMAS ISLAND
CY	CYPRUS
DD	GERMAN DEMOCRATIC REPUBLIC
DE	FEDERAL REPUBLIC OF GERMANY
DK	DENMARK
DM	DOMINICA
DO	DOMINICAN REPUBLIC
DZ	ALGERIA
EC	ECUADOR
EG	EGYPT
EH	WESTERN SAHARA
EP	EUROPEAN PATENT OFFICE
ES	SPAIN
ET	ETHIOPIA
FI	FINLAND
FJ	FIJI
FK	FALKLAND ISLANDS
FO	FAROE ISLANDS
FQ	FRENCH SOUTHERN AND ANTARTIC ISLANDS
FR	FRANCE
GA	GABON
GB	UNITED KINGDOM
GD	GRENADA
GE	GILBERT ISLANDS
GF	FRENCH GUIANA
GH	GHANA
GI	GIBRALTAR
GL	GREENLAND
GM	GAMBIA
GN	GUINEA
GP	GUADALOUPE
GQ	EQUATORIAL GUINEA
GR	GREECE
GT	GUATEMALA
GU	GUAM
GW	GUINEA-BISSAU
GY	GUYANA
HK	HONG KONG
HM	HEARD AND MCDONALD ISLANDS
HN	HONDURAS
HT	HAITI
HU	HUNGARY
HV	UPPER VOLTA
ID	INDONESIA
IE	IRELAND
IL	ISRAEL
IN	INDIA

IO	BRITISH INDIAN OCEAN TERRITORY
IQ	IRAQ
IR	IRAN
IS	ICELAND
IT	ITALY
JM	JAMAICA
JO	JORDAN
JP	JAPAN
JT	JOHNSTON ATOLL
KE	KENYA
KH	KAMPUCHEA
KM	COMOROS
KN	ST KITTS-NEVIS-ANGUILLA
KP	DEMOCRATIC PEOPLE'S REPUBLIC OF KOREA
KR	REPUBLIC OF KOREA
KW	KUWAIT
KY	CAYMAN ISLANDS
LA	LAOS
LB	LEBANON
LC	ST LUCIA
LI	LIECHTENSTEIN
LK	SRI LANKA
LR	LIBERIA
LS	LESOTHO
LU	LUXEMBOURG
LY	LIBYA
MA	MOROCCO
MC	MONACO
MG	MADAGASCAR
MI	MIDWAY ISLANDS
ML	MALI
MN	MONGOLIA
MO	MACAO
MQ	MARTINIQUE
MR	MAURITANIA
MS	MONTSERRAT
MT	MALTA
MU	MAURITIUS
MV	MALDIVES
MW	MALAWI
MX	MEXICO
MY	MALAYSIA
MZ	MOZAMBIQUE
NA	NAMIBIA
NC	NEW CALEDONIA
NE	NIGER
NF	NORFOLK ISLAND
NG	NIGERIA
NH	NEW HERBRIDES
NI	NICARAGUA
NL	NETHERLANDS
NO	NORWAY
NP	NEPAL
NR	NAURU
NT	IRAQ-SAUDI NEUTRAL
NU	NIUE
NV	NAVASSA ISLAND
NZ	NEW ZEALAND
OA	AFRICAN INTELLECTUAL PROPERTY ORGANIZATION
OM	OMAN
PA	PANAMA
PC	TRUST TERRITORY OF THE PACIFIC ISLANDS
PE	PERU
PF	FRENCH POLYNESIA
PG	PAPUA NEW GUINEA
PH	PHILIPPINES
PI	PARACEL ISLANDS
PK	PAKISTAN
PL	POLAND
PM	ST PIERRE AND MIQUELON
PN	PITCAIRN
PR	PUERTO RICO
PT	PORTUGAL
PU	US MISC, PACIFIC ISLANDS
PY	PARAGUAY
PZ	CANAL ZONE
QA	QATAR
RE	REUNION
RH	ZIMBABWE
RO	ROMANIA
RW	RWANDA
SA	SAUDI ARABIA
SB	SOLOMON ISLANDS
SC	SEYCHELLES
SD	SUDAN
SE	SWEDEN
SG	SINGAPORE
SH	ST HELENA
SI	SPRATLY ISLANDS
SJ	SVALBARD AND JAN MAYEN
SL	SIERRA LEONE
SM	SAN MARINO
SN	SENEGAL
SO	SOMALIA
SR	SURINAM
ST	SAO TOME AND PRINCIPE
SU	UNION OF SOVIET SOCIALIST REPUBLICS
SV	EL SALVADOR
SY	SYRIA
SZ	SWAZILAND
TC	TURKS AND CAICOS ISLANDS

TD	CHAD
TG	TOGO
TH	THAILAND
TK	TOKOLAU ISLANDS
TN	TUNISIA
TO	TONGA
TR	TURKEY
TT	TRINIDAD AND TOBAGO
TV	TUVALU
TW	TAIWAN
TZ	UNITED REPUBLIC OF TANZANIA
UG	UGANDA
US	UNITED STATES
UY	URUGUAY
VA	VATICAN CITY
VC	ST VINCENT

VE	VENEZUELA
VG	BRITISH VIRGIN ISLANDS
VI	VIRGIN ISLANDS OF THE US
VN	VIETNAM
VS	SOUTH VIETNAM
WF	WALLIS AND FUTUNA
WK	WAKE ISLAND
WO	WORLD INTELLECTUAL PROPERTY ORGANIZATION
WS	WESTERN SAMOA
YD	YEMEN (ADEN)
YE	YEMEN (SANA)
YU	YUGOSLAVIA
ZA	SOUTH AFRICA
ZM	ZAMBIA
ZR	ZAIRE

B.2 COUNTRY ORDER

AI	AFARS AND ISSAS
AF	AFGHANISTAN
OA	AFRICAN INTELLECTUAL PROPERTY ORGANIZATION
AL	ALBANIA
DZ	ALGERIA
AS	AMERICAN SAMOA
AD	ANDORRA
AO	ANGOLA
AQ	ANTARCTICA
AG	ANTIGUA
AR	ARGENTINA
AU	AUSTRALIA
AT	AUSTRIA
BS	BAHAMAS
BH	BAHRAIN
BD	BANGLADESH
BB	BARBADOS
BE	BELGIUM
BZ	BELIZE
BJ	BENIN
BM	BERMUDA
BT	BHUTAN
BO	BOLIVIA
BW	BOTSWANA
BV	BOUVETOYA
BR	BRAZIL
IO	BRITISH INDIAN OCEAN TERRITORY
VG	BRITISH VIRGIN ISLANDS
BN	BRUNEI
BG	BULGARIA

BU	BURMA
BI	BURUNDI
CM	CAMEROON
CA	CANADA
PZ	CANAL ZONE
CT	CANTON AND ENDERBURG ISLANDS
CV	CAPE VERDE
KY	CAYMAN ISLANDS
CF	CENTRAL AFRICAN REPUBLIC
TD	CHAD
CL	CHILE
CN	CHINA
CX	CHRISTMAS ISLAND
CC	COCOS ISLANDS
CO	COLOMBIA
KM	COMOROS
CG	CONGO
CK	COOK ISLAND
CR	COSTA RICA
CU	CUBA
CY	CYPRUS
CS	CZECHOSLOVAKIA
KP	DEMOCRATIC PEOPLE'S REPUBLIC OF KOREA
DK	DENMARK
DM	DOMINICA
DO	DOMINICAN REPUBLIC
EC	ECUADOR
EG	EGYPT
SV	EL SALVADOR
GQ	EQUATORIAL GUINEA
ET	ETHIOPIA

EP	EUROPEAN PATENT OFFICE
FK	FALKLAND ISLANDS
FO	FAROE ISLANDS
DE	FEDERAL REPUBLIC OF GERMANY
FJ	FIJI
FI	FINLAND
FR	FRANCE
GF	FRENCH GUIANA
PF	FRENCH POLYNESIA
EQ	FRENCH SOUTHERN AND ANTARCTIC ISLANDS
GA	GABON
GM	GAMBIA
DD	GERMAN DEMOCRATIC REPUBLIC
GH	GHANA
GI	GIBRALTAR
GE	GILBERT ISLANDS
GR	GREECE
GL	GREENLAND
GD	GRENADA
GP	GUADALOUPE
GU	GUAM
GT	GUATEMALA
GN	GUINEA
GW	GUINEA-BISSAU
GY	GUYANA
HT	HAITI
HM	HEARD AND MCDONALD ISLANDS
HN	HONDURAS
HK	HONG KONG
HU	HUNGARY
IS	ICELAND
IN	INDIA
ID	INDONESIA
IR	IRAN
IQ	IRAQ
NT	IRAQ-SAUDI NEUTRAL ZONE
IE	IRELAND
IL	ISRAEL
IT	ITALY
CI	IVORY COAST
JM	JAMAICA
JP	JAPAN
JT	JOHNSTON ATOLL
JO	JORDAN
KH	KAMPUCHEA
KE	KENYA
KW	KUWAIT
LA	LAOS
LB	LEBANON
LS	LESOTHO
LR	LIBERIA
LY	LIBYA
LI	LIECHTENSTEIN
LU	LUXEMBOURG
MO	MACAO
MG	MADAGASCAR
MW	MALAWI
MY	MALAYSIA
MV	MALDIVES
ML	MALI
MT	MALTA
MQ	MARTINIQUE
MR	MAURITANIA
MU	MAURITIUS
MX	MEXICO
MI	MIDWAY ISLANDS
MC	MONACO
MN	MONGOLIA
MS	MONTSERRAT
MA	MOROCCO
MZ	MOZAMBIQUE
NA	NAMIBIA
NR	NAURU
NV	NAVASSA ISLAND
NP	NEPAL
AN	NETHERLAND ANTILLES
NL	NETHERLANDS
NC	NEW CALEDONIA
NH	NEW HEBRIDES
NZ	NEW ZEALAND
NI	NICARAGUA
NE	NIGER
NG	NIGERIA
NU	NIUE
NF	NORFOLK ISLAND
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PY	PARAGUAY
PE	PERU
PH	PHILIPPINES
PN	PITCAIRN
PL	POLAND
PT	PORTUGAL
PR	PUERTO RICO
QA	QATAR
RE	REUNION
KR	REPUBLIC OF KOREA
RO	ROMANIA
RW	RWANDA
SM	SAN MARINO
ST	SAO TOME AND PRINCIPE

SA	SAUDI ARABIA
SN	SENEGAL
SC	SEYCHELLES
SL	SIERRA LEONE
SG	SINGAPORE
SB	SOLOMON ISLANDS
SO	SOMALIA
ZA	SOUTH AFRICA
VS	SOUTH VIETNAM
ES	SPAIN
SI	SPRATLY ISLANDS
LK	SRI LANKA
SH	ST HELENA
KN	ST KITTS-NEVIS-ANQUILLA
LC	ST LUCIA
PM	ST PIERRE AND MIQUELON
VC	ST VINCENT
SD	SUDAN
SR	SURINAM
SJ	SVALBARD AND JAN MAYEN
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SE	SWEDEN
CH	SWITZERLAND
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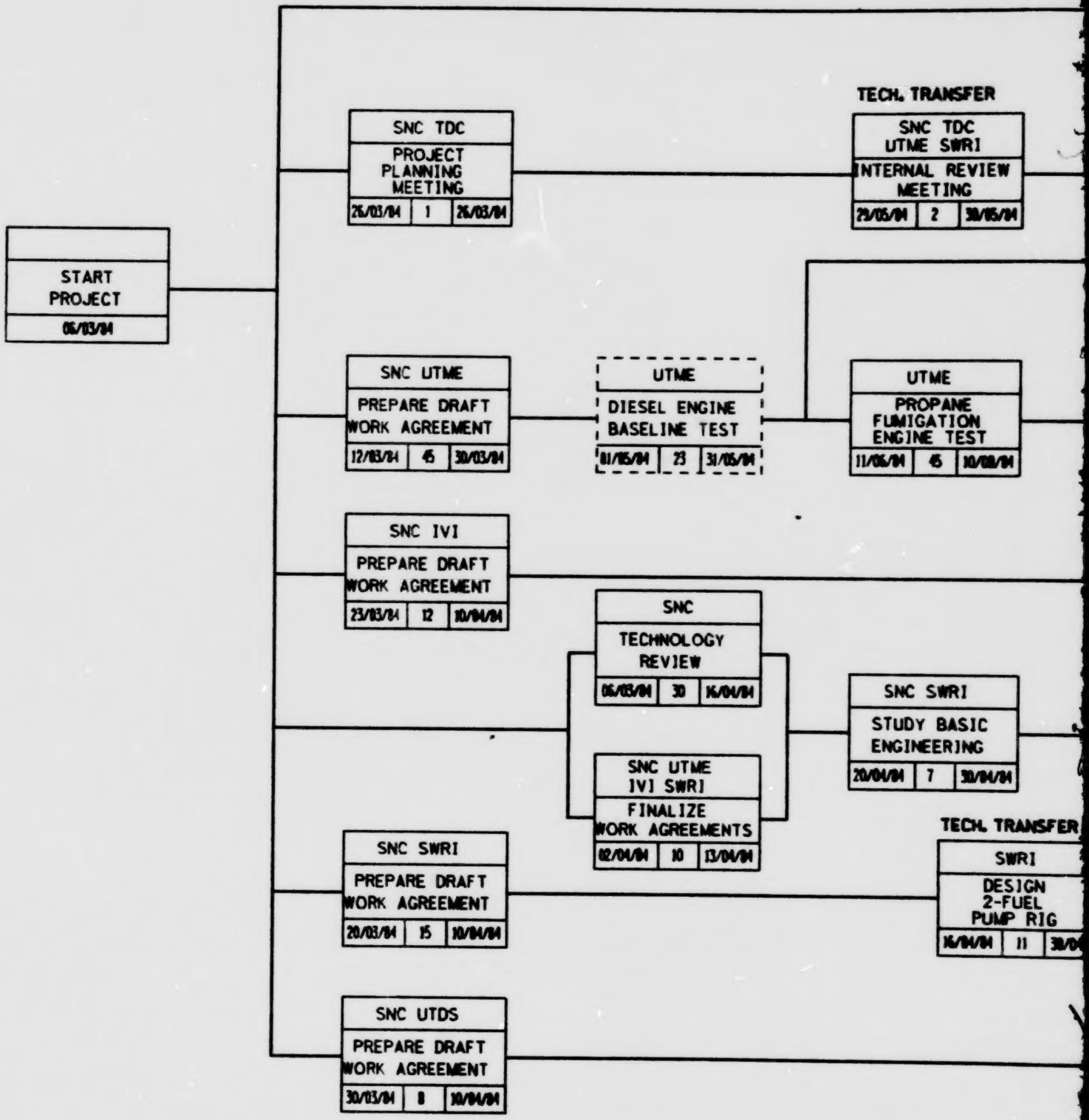
TW	TAIWAN
TH	THAILAND
TG	TOGO
TK	TOKOLAU ISLANDS
TO	TONGA
TT	TRINIDAD AND TOGABO
PC	TRUST TERRITORY OF THE PACIFIC ISLANDS
TN	TUNISIA
TR	TURKEY
TC	TURKS AND CAICOS ISLANDS
TV	TUVALU
UG	UGANDA
SU	UNION OF SOVIET SOCIALIST REPUBLICS
AE	UNITED ARAB EMIRATES
GB	UNITED KINGDOM
TZ	UNITED REPUBLIC OF TANZANIA
US	UNITED STATES
HV	UPPER VOLTA
UY	URUGUAY
PU	US MISC. PACIFIC ISLANDS
VA	VATICAN CITY
VE	VENEZUELA
VN	VIETNAM
VI	VIRGIN ISLANDS OF THE US
WK	WAKE ISLAND
WF	WALLIS AND FUTUNA
EH	WESTERN SAHARA
WS	WESTERN SAMOA
WO	WORLD INTELLECTUAL PROPERTY ORGANIZATION
YD	YEMEN (ADEN)
YE	YEMEN (SANA)
YU	YUGOSLAVIA
ZR	ZAIRE
ZM	ZAMBIA
RH	ZIMBABWE

APPENDIX C

DRAWINGS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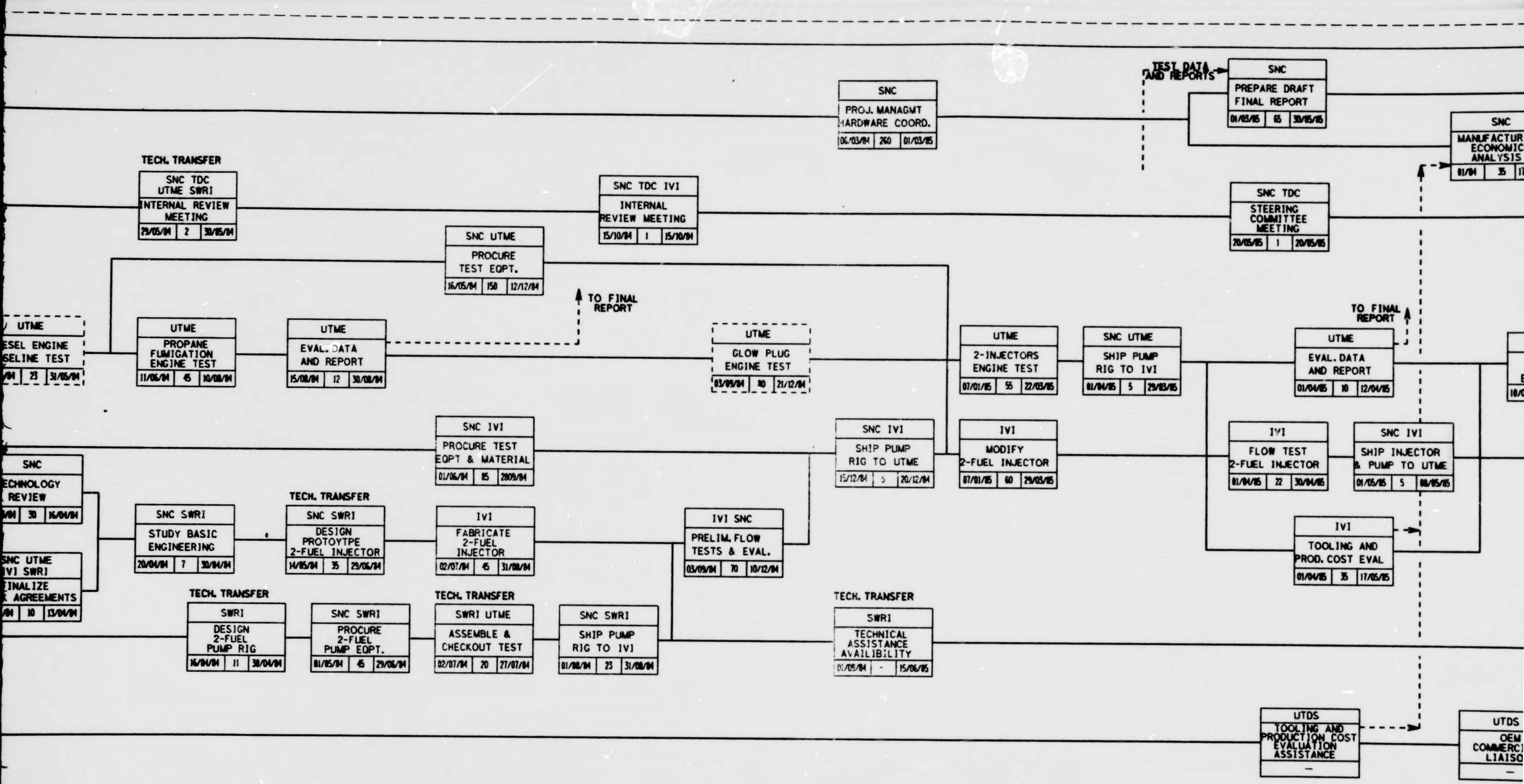

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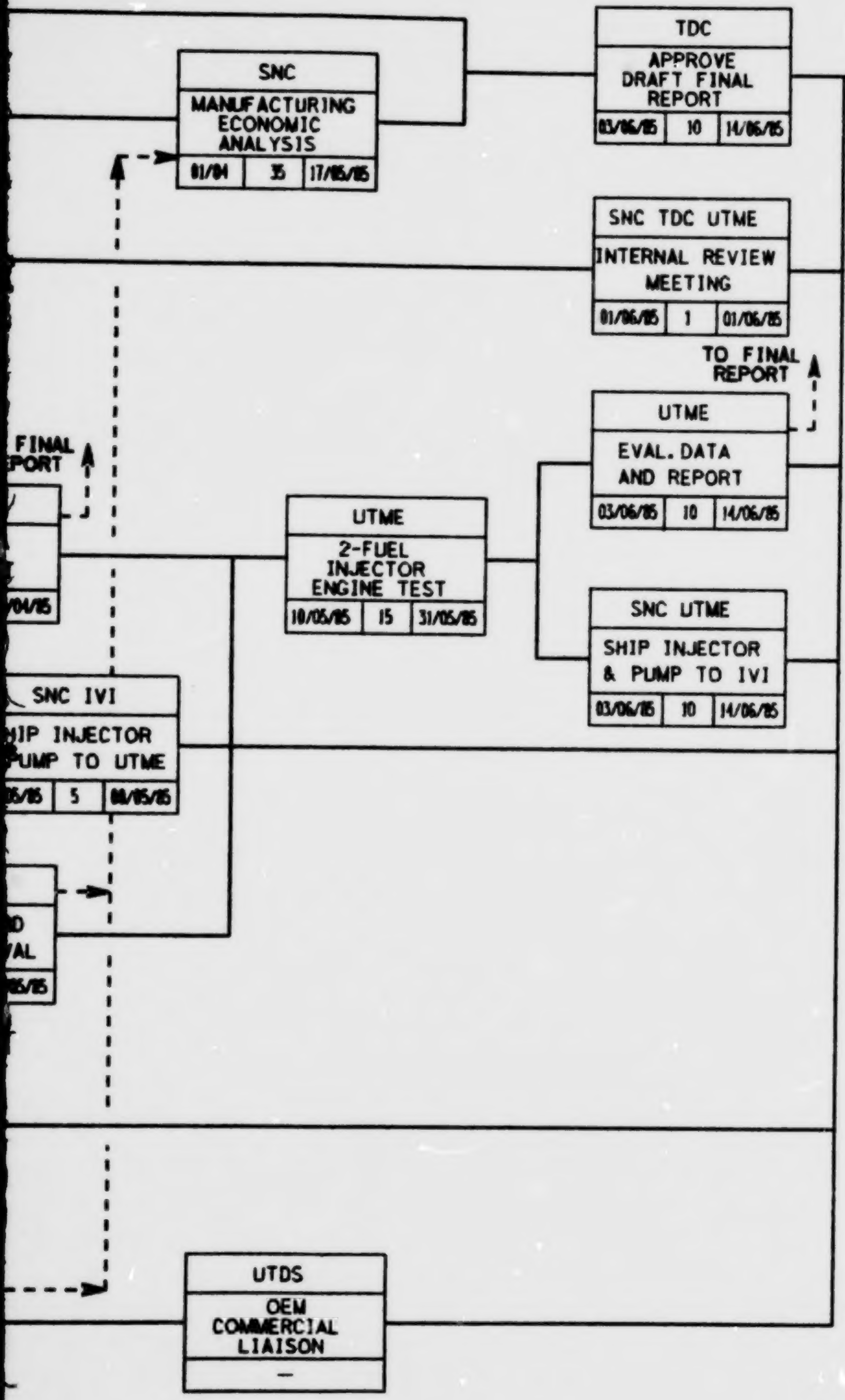
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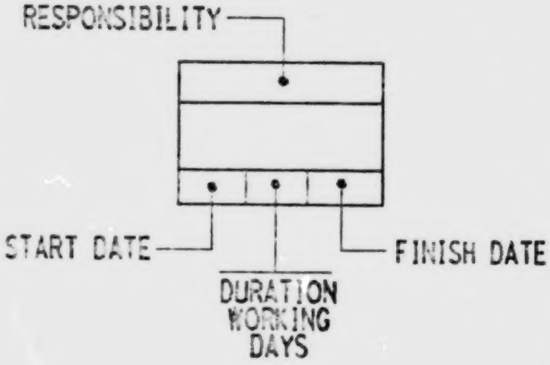
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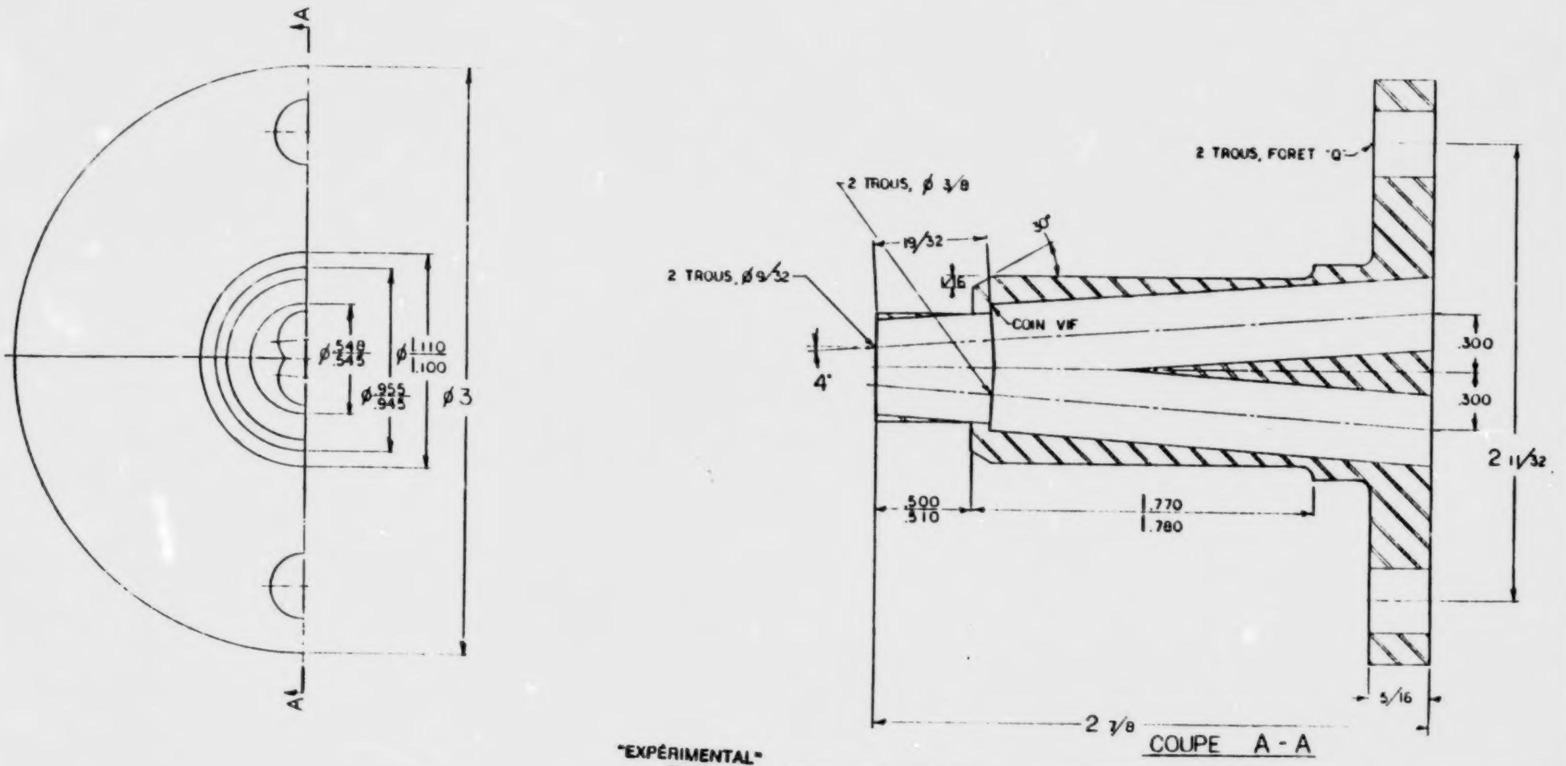
START DATE
DURATION WORKING DAYS
FINISH DATE

SNC= SNC Inc.
IVI= IVI Inc.
UTME= UNIVERSITY OF TORONTO
DEPT. OF MECHANICAL ENG.
SWRI= SOUTHWEST RESEARCH INSTITUTE
UTDS= UNITED TECHNOLOGIES DIESEL SYSTEMS

CLIENT TDC PROJ. PROPANE/DIESEL INJECTION SYSTEM	FAIT / MADE P.N.	VÉRIFIÉ / CHKD P.N.	DATE 30-04-84
	APPR.	ÉCH. / SCALE	DATE REV.
SNC — SPEC NETWORK DIAGRAM	CONTR. 8065 / SUBDIV.	ELEMENT	NO. / REV. 0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ACIER DOUX

LES INDUSTRIES VALCARTIER INC

SUPPORT D'INJECTEUR

INJECTEUR

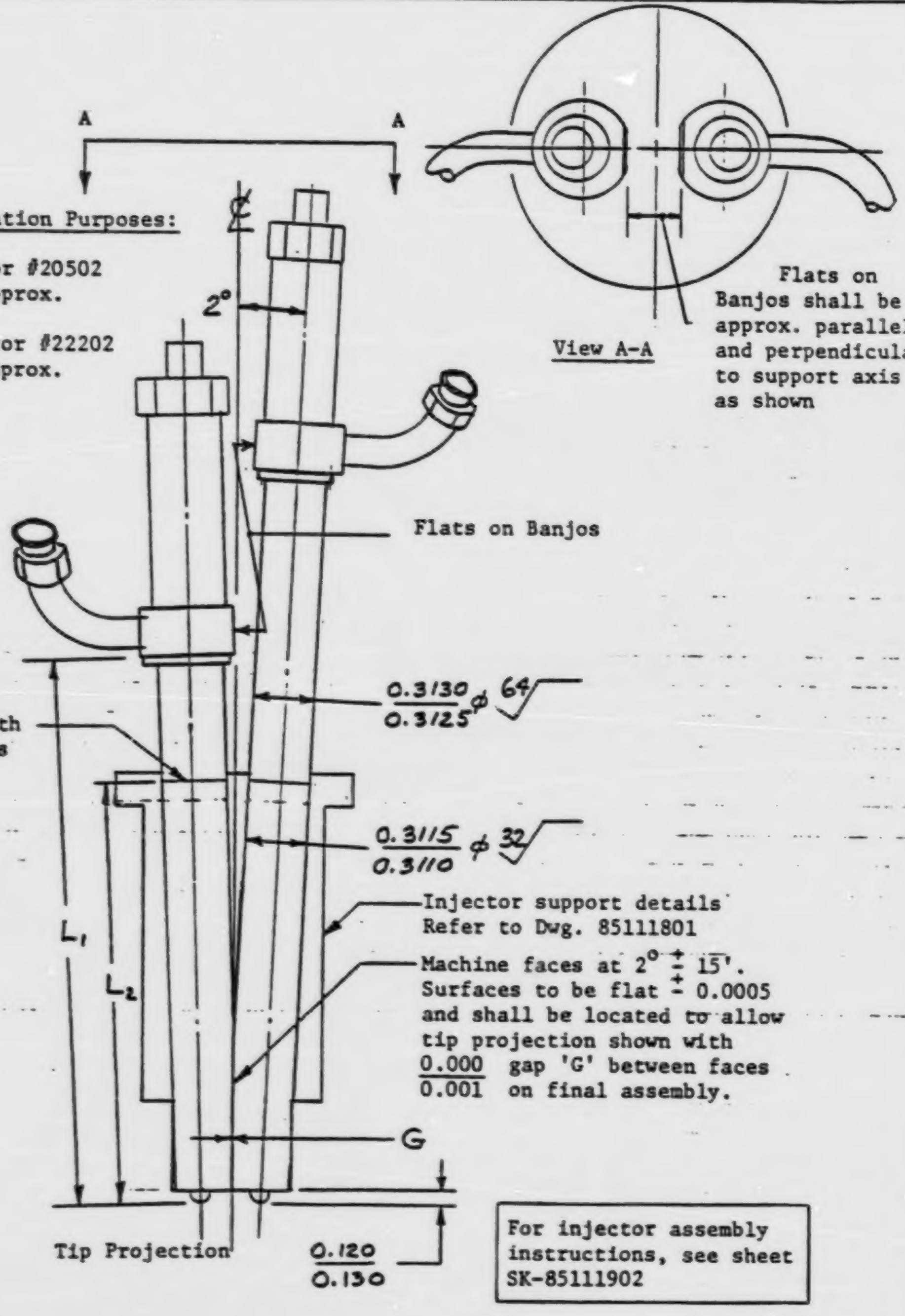
23-1-85

2X

RD-4-020

snc	DOSSIER D'ÉTUDE DESIGN BRIEF	Client Transport Canada	Contr	Code du dossier d'étude Design brief code	Rev. N°/No.	Page
		Projet/Project 2-Fuel Injector	7325	4 EB		

Sujet/Subject Dwg. SK-85111901 Injectors Machining and Assembly	Prepare par/Prepared by P. Nikolaiczuk	Date 85.11.19
	Verifie par/Checked by	Date



For Identification Purposes:

Diesel Injector #20502
L_1 = 4.00 Approx.

Propane Injector #22202
L_1 = 3.25 Approx.

Flats on Banjos shall be approx. parallel and perpendicular to support axis as shown

Flats on Banjos

Shoulder on both injector bodies at L_2 = 2.80

Injector support details Refer to Dwg. 85111801

Machine faces at 2° ± 15'. Surfaces to be flat ± 0.0005 and shall be located to allow tip projection shown with $\frac{0.000}{0.001}$ gap 'G' between faces on final assembly.

Tip Projection

For injector assembly instructions, see sheet SK-85111902

APPENDIX D

UTME TEST REPORT

PROPANE-FUELED DIESEL ENGINES

Final Report

Submitted to

SNC, Inc.

by

U. Oester, J.S. Wallace, L. Segal and J.F. Keffer

Engine Research and Development Laboratory
Department of Mechanical Engineering
University of Toronto

Revision A

June 1986

EXECUTIVE SUMMARY

This project used a single-cylinder prechamber diesel engine to investigate the use of propane as a diesel engine fuel. Two methods of delivering fuel to the engine were evaluated. In both cases, pilot injection of a small quantity of diesel fuel was used to provide ignition, since propane has a very low cetane number. In the first method, fumigation, gaseous propane at ambient pressure was supplied to the engine's inlet manifold, where it was mixed with air entering the cylinder. In the second method, liquid propane was pumped to a high pressure and injected directly into the engine's prechamber at the same time as the diesel pilot was injected. Separate pumps and injectors were used for the propane and diesel fuels, although the two pencil injectors were mounted in a single holder which fit the mounting boss provided for the original Bosch diesel fuel injector on the cylinder head. These were standard production pencil injectors intended for use in other engines so their spray patterns were not optimized for this combustion chamber.

Tests were carried out at three compression ratios, 19:1, 22:1 and 25:1 and at three engine speeds, 1200, 1500 and 1800 rpm, first with the engine in its original diesel form and subsequently with the two propane fueling methods (fumigation and dual fueling).

The results showed that both propane fueling methods resulted in significantly less exhaust smoke than the original diesel at high loads. For the same level of smoke opacity, the dual-fueled propane diesel produced about 16% more power. The

power output of the fumigated engine was knock limited to about the same level as that of the original diesel.

The dual-fueled propane diesel allowed greater substitution of propane for diesel fuel. At the compression ratios and speeds tested, only 10-15% of the total fuel energy had to be provided by the diesel pilot injection to maintain good ignition and stable operation. The amount of propane that could replace diesel fuel in the fumigated engine was limited by the onset of knock and decreased as the compression ratio increased. At 22:1, as much as 25% of the total fuel energy had to be provided by the diesel pilot injection to prevent the rate of pressure rise from exceeding 345 kPa/degree of crankshaft rotation (50 psi/degree). Rates of pressure rise greater than this will result in excessive engine vibration and noise. It should be noted that even larger amounts of diesel fuel would be required in a fumigated direct injection engine. The high rate of mixing and subsequent rapid burning in the prechamber of this test engine reduce the susceptability of this particular engine to knock.

With the exception of the tests at 1200 rpm for the 22:1 compression ratio, the energy consumption of the dual-fuel injection engine was nearly the same as that of the original diesel. In general, the energy consumption is low at light loads, increasing at high loads as smoke begins to form. The total energy consumption of the propane fumigated engine falls with increasing engine load until it is equal to that of the original diesel at full load.

NO_x emissions are lower with dual fueling than with the original diesel. No significant effect of compression ratio or speed was evident. NO_x emissions of the propane fumigated engine were very low at light engine loads, but increased sharply as the load increased.

The dual-fuel engine was also much more stable and easier to control than the propane fumigated engine throughout the operating range.

The tradeoff for the advantages of dual-fueling is the increased mechanical complexity of the system. Additional development is needed to provide a simpler and more reliable engine fueling system. A number of problem areas have been identified in this initial project which should be addressed in subsequent work. However, the results obtained are quite promising, particularly since no optimization of the injection system was carried out for this phase of the work. The project has clearly demonstrated the feasibility and potential benefits of the dual-fuel propane diesel engine.

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1. INTRODUCTION

Propane has been the most successful of potential alternative fuels for transportation in Canada. Its primary attraction is reduced cost (due to the absence of motor fuel taxes), but unlike most other alternative fuels, vehicle range and drivability with propane are nearly equivalent to that with gasoline.

With increasing production of natural gas in Canada, the supply of propane should increase modestly as well. This increase will be partially offset by the loss of propane output from inactivated oil refineries. While the amount of propane produced will never be able to meet all of Canada's transportation fuel needs, a steadily growing supply will enable propane to be substituted for gasoline in a greater number of applications.

Propane is widely distributed geographically, both in motor vehicle quality and in commercial quality for non-engine applications. The number of vehicle refueling stations in a given area is still small, however. Thus, propane is best suited for use in fleet vehicles which typically are refueled at a central facility. Equipment to convert the spark-ignition gasoline-fueled engines in light and medium duty vehicles to use propane is widely available, although investigations carried out for the Ontario Ministry of Transportation and Communications (MTC) by the Engine Research and Development Laboratory (ERDL) of the University of Toronto in the Department of Mechanical Engineering (UTME) have shown that currently available equipment does not provide optimum performance and economy (1). Despite the availability of this equipment, there is still a sizeable segment of the total fleet market which cannot presently be converted to propane. Virtually all heavy-duty vehicles and an increasing number of medium and light duty vehicles are powered by diesel engines, for which no propane conversion equipment is available. The research work performed in this project has demonstrated the feasability of successfully operating diesel cycle engines on propane.

There are three potential methods of using propane in diesel engines. In the first method, propane is mixed with air and the resulting mixture introduced into the engine cylinders, a technique known as fumigation. Ignition is provided by the injection of a small quantity of diesel fuel in the normal manner (pilot injection). The engine is therefore a dual-fuel engine. The proportions of propane and diesel fuel must be carefully controlled if excessive rates of pressure rise during the combustion process are to be avoided (2,3).

Fumigation is effective in constant speed engine applications such as generating sets, because the engine's governor decreases the amount of diesel fuel injected to compensate for the propane inducted with the air. Fumigated vehicle engines have a mechanical linkage to control the ratio between propane and diesel fuel. This arrangement is a compromise, however, as it provides a constant percentage of propane while the amount of propane that can be tolerated by the engine varies with operating conditions. In addition, the linkage may be tampered with to provide more fuel (and therefore more power), to the detriment of engine durability and pollutant emission levels. Finally, the amount of propane that can be substituted for diesel fuel is limited to about 30-40% in most fumigated engines. Thus, despite its simplicity, propane fumigation is not a particularly effective fueling method for transportation applications.

Another dual-fuel possibility is to inject liquid propane into the engine, along with a small pilot injection of diesel fuel to provide ignition. The ignition source is necessary because propane has a very low cetane number. The use of liquid propane injection and diesel pilot injection in a diesel engine has been evaluated in a cooperative investigation carried out by UTME and SNC, Inc. of Montreal. This document presents the results of UTME's part of the work.

A third possible method of using propane in diesel engines is to inject liquid propane into the engine and to use a continuously operating glow plug to provide ignition. This method has the advantage of allowing total substitution of propane for diesel fuel. A separate investigation of this method was carried out by UTME for Transportation Development Centre (4).

Because propane is stored as a compressed liquid, significant modifications to the fuel supply system and the fuel injection pump and injector were necessary to keep the propane liquified and to accommodate propane's low lubricity. The following sections of this report detail the modifications made and present the results of an extensive test program to evaluate the feasability of operating diesel engines on injected propane with diesel pilot injection.

2. EXPERIMENTAL FACILITIES

2.1 TEST ENGINE

"WAUKESHA" CFR (Cooperative Fuel Research) one-cylinder variable compression indirect injection (IDI) diesel S/N 791712.

Bore: 3.25 in (82.55 mm.)
Stroke: 4.50 in (114.30 mm.)
Displacement: 37.33 in^3 (611.73 cm^3)
Antechamber: swirl combustion type
  bore: 1.625 in (41.275 mm.)
  length: adjustable

Figure 2-1 shows the variable compression head and the cylinder of the CFR engine. Compression ratio is varied by screwing an expansible plug in or out of the cylindrical combustion chamber. For this project, the variable compression ratio plug was replaced with a plug of fixed dimensions incorporating a mounting hole for a pressure transducer (Figure 2-2). Compression ratios of 19:1, 22:1 and 25:1 could be selected by fitting shims between the plug and the mounting face of the cylinder head. This freed the threaded access hole at the top of the combustion chamber, normally used to mount a knock sensor, for installation of a glow plug and/or injector.

Three different fuel delivery arrangements were used. For baseline tests with diesel fuel, the American Bosch injection pump and injector standard on the CFR cetane engine were used. This system also served to deliver the pilot injection of diesel fuel during the fumigation tests. Gaseous propane was supplied to the engine during the fumigation tests through a 1/4 inch diameter tube drilled with opposed holes and placed across the air inlet pipe. A two stage regulator-vaporizer and a single stage final regulator reduced the propane pressure to approximately 2 kPa (0.3 psi).

The fuel system was changed completely for the tests with liquid propane injection. Due to the fact that propane is a gas at atmospheric temperature and pressure, fuel system components

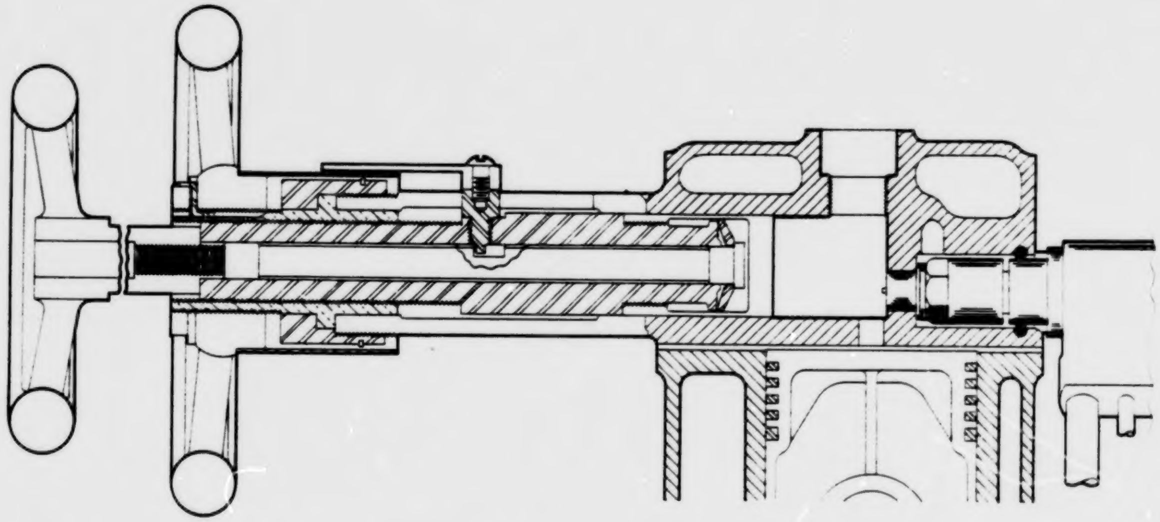


Figure 2-1 Variable compression head and cylinder of CFR cetane engine.

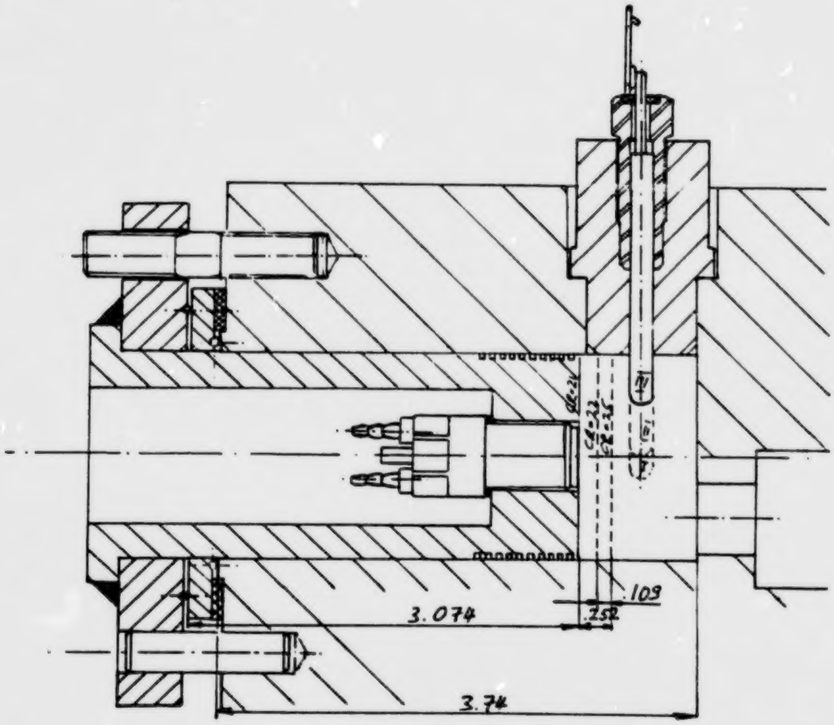
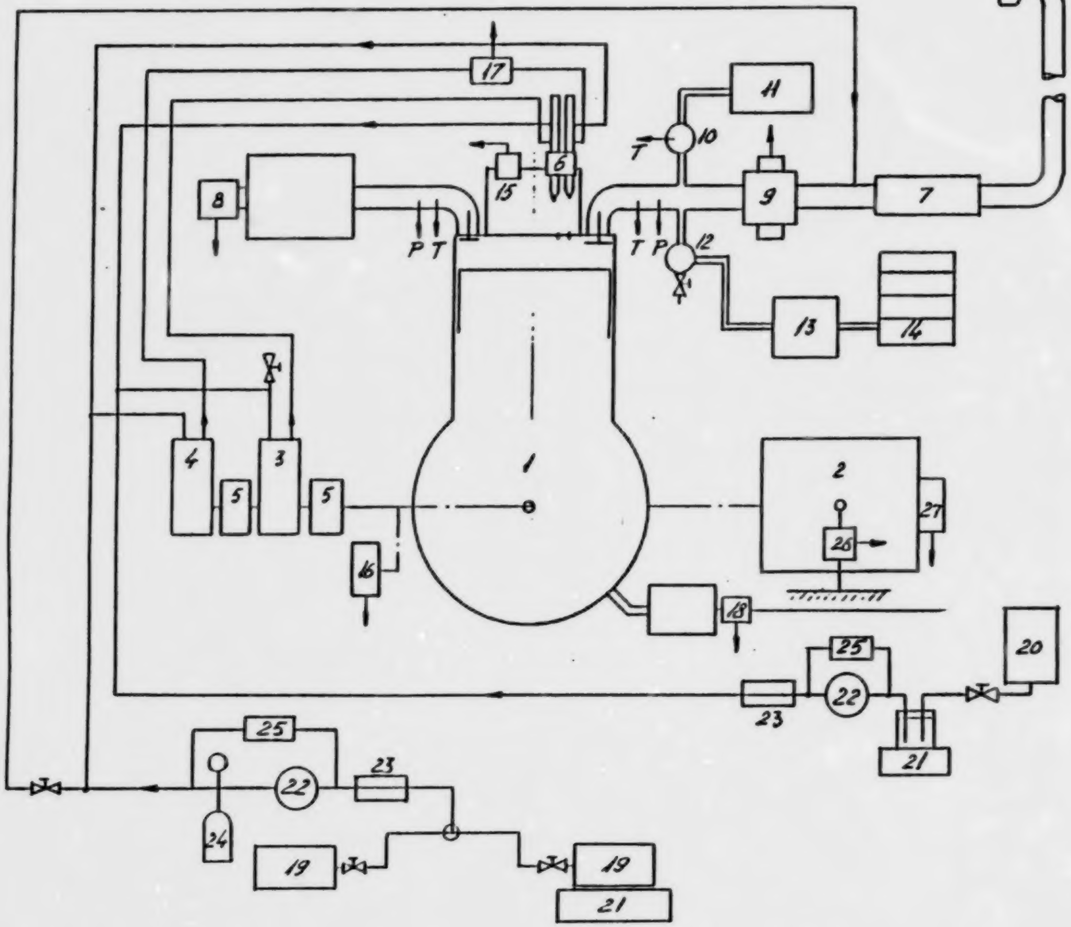


Figure 2-2 Fixed compression ratio swirl chamber plug with pressure transducer.

snc



1 - engine
2 - dynamometer
3 - diesel injection pump
4 - liquid propane injection pump
5 - angular phase shifting mechanism
6 - double injector
7 - muffler
8 - inlet air flowmeter and pulsation damper
9 - smokemeter
10 - heated filter and lines
11 - heated HC analzyer
12 - filter and drain
13 - exhaust sample conditioner
14 - NO, O_2, CO and CO_2 analyzers
15 - cylinder pressure transducer
16 - crank angle encoder
17 - propane line pressure transducer
18 - blow-by flowmeter and pulsation damper
19 - propane cylinders
20 - diesel supply tank
21 - balance
22 - pump
23 - filter
24 - accummulator
25 - pressure check valve
26 - load cell
27 - tachometer
T - temperature measuring point
P - pressure measuring point

Figure 2-3. Overall layout of test equipment and instrumentation.

in addition to those normally present on a diesel engine were necessary to keep the propane liquified. A block diagram of these components is included in Figure 2-3. Propane from the liquid takeoff of the tank was first filtered using a Matheson 6184 replaceable element high pressure filter (100% efficient to 0.2 microns) to remove dirt and debris (Extensive laboratory test experience with propane has shown that even HD-5 propane is very dirty - excellent filters are a must). A diaphragm pump (American Lewa model EKM-1 with 16 mm size M510S head), driven by an explosion proof 110 VAC electric motor was used to pressurize the liquid propane drawn from the tank to 4.2 to 5.2 bars (600-750 psi). This pressure is above the saturation pressure for any temperature likely encountered in the fuel supply system and should therefore ensure that the propane stays liquified. The pressure is maintained by a relief valve (Nupro 6S 4R3A) which recirculates propane as necessary to the pump inlet. An accumulator (Greerolater 800 201) was used to dampen pulses from the supply pump. Figure 2-4 shows the primary pressure supply system.

The standard fuel injection pump was replaced by two Robert Bosch fuel injection pumps (PFR 1K80A 427/11), one for the pilot injection of diesel fuel and one for liquid propane injection. The camshafts which actuated these pumps were connected in series and driven from the front end of the engine camshaft as shown in Figure 2-5. An American Bosch Model TMB 12A2 variable timing device was incorporated between the drive and the first pump (diesel fuel) to allow setting the diesel pilot injection timing. A Candy Manufacturing Company Model 1A Dynamic Differential was used to drive the second pump (liquid propane) from the first so that the phase angle between the two pumps could be varied. This allowed setting the liquid propane injection timing to a different value than the diesel pilot injection. Pressurized propane from the primary supply system was fed to the second pump. Extensive modifications to the barrel and plunger (6 mm diameter #1418405002-800) were necessary to ensure the delivery of propane with minimum leakage and minimum wear despite the low

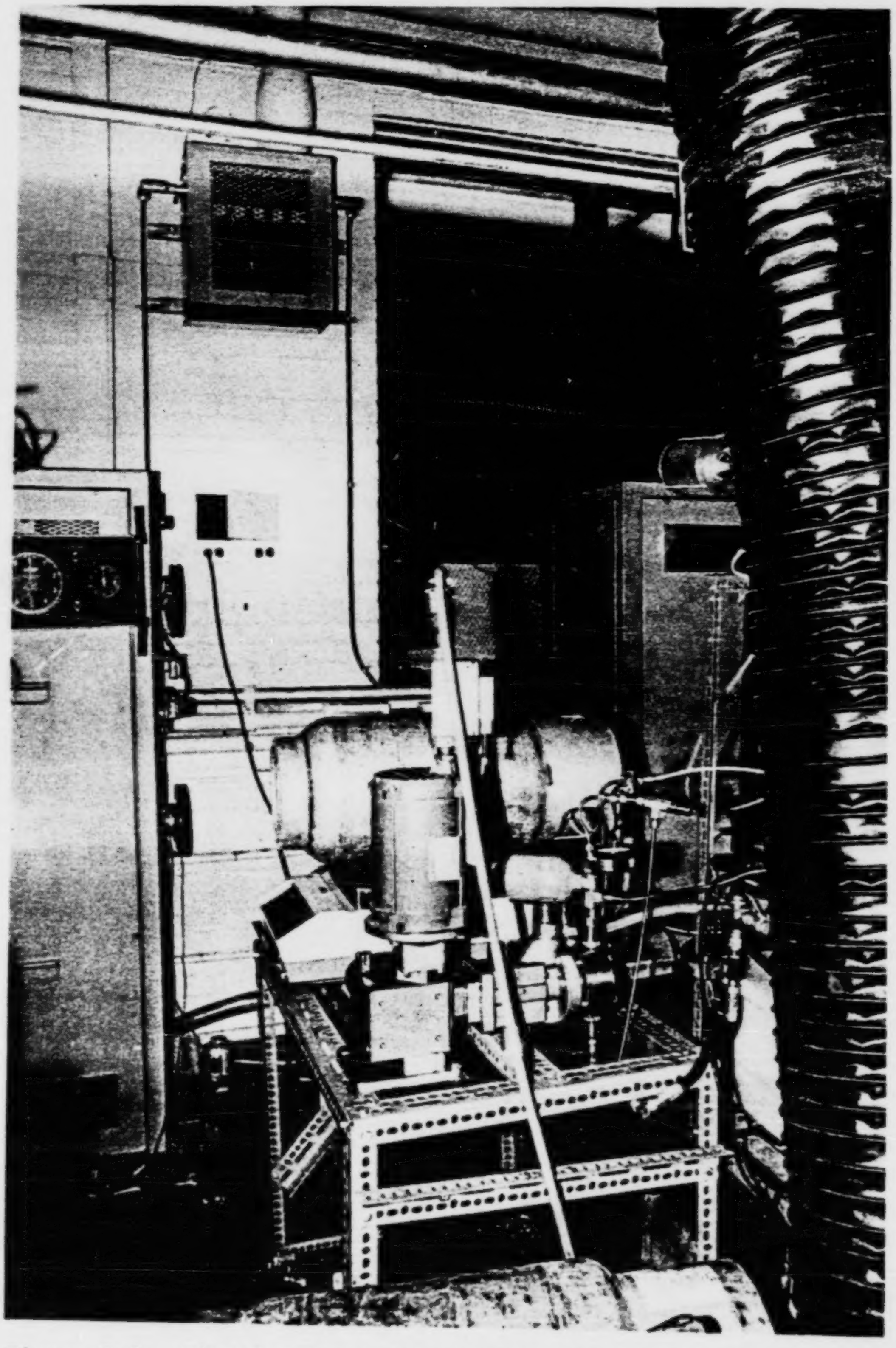


Figure 2-4. Liquid Propane Consumption Measurement and Primary Supply Systems.

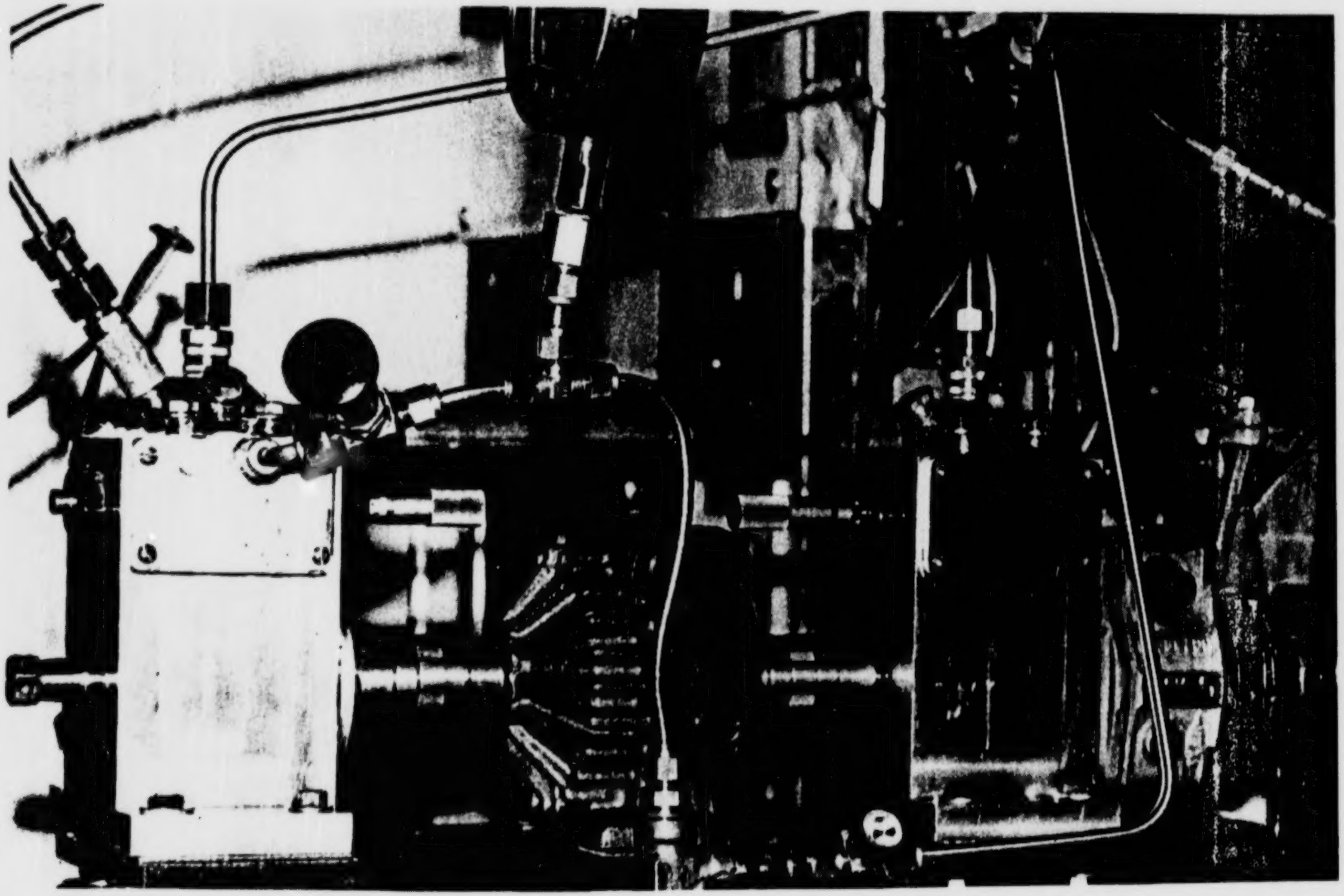
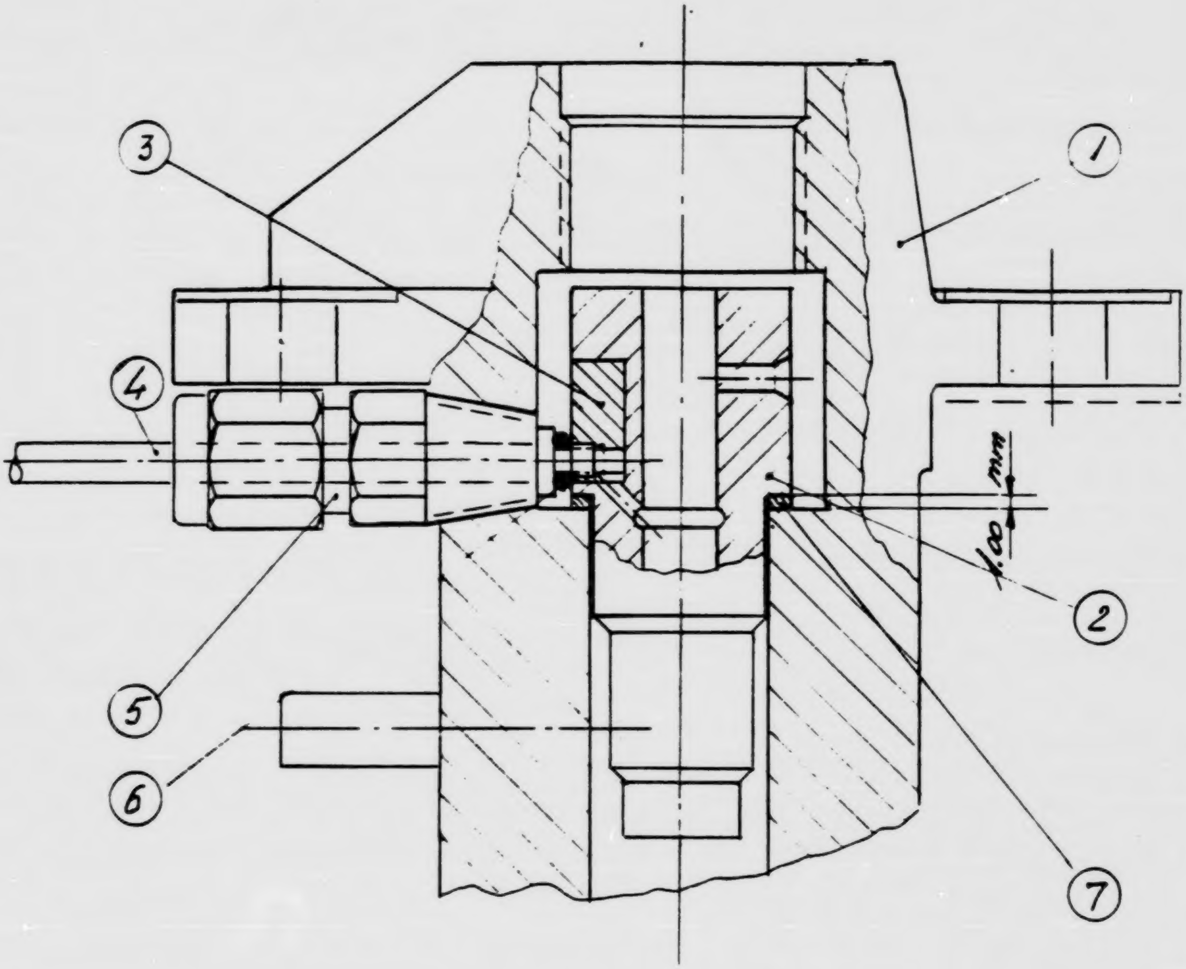


Figure 2-5. Detail of Liquid Propane and Diesel Fuel Injection Pumps. Note phase control systems, micrometer barrels for setting pump rack position and supplementary lubrication on the propane pump.

lubricity of propane compared to diesel fuel. Referring to Figure 2-6, a copper gasket was added under the barrel shoulder to avoid leaks from the fuel supply chamber to the lower housing containing the camshaft and roller tappet. This was necessary because the fuel inlet pressure is nearly 100 times that used with diesel fuel. The groove and inclined channel built into the barrel to return the diesel fuel high-pressure leak-off to the supply side was modified to supply lubrication oil to the barrel and plunger assembly. The barrel was machined to accept a mild steel insert (#3 in Figure 2-6) containing a 1/8 inch diameter oil supply tube (#4). The insert was glued to the barrel with epoxy to avoid welding or soldering. Heat from either of the latter would have destroyed the precision fit of the plunger and barrel assembly. The oil supply tube was held in place by a Swagelok male connector (#5). The lower end of the connector compressed an o-ring (#6) which sealed the oil supply tube to the barrel.

Oil for sealing and lubrication was provided by a cylinder pressurized by propane at 4.1-5.6 bars (600-750 psi) from the supply pump (This system is shown in Figure 2-5 but not 2-3). A check valve was necessary to prevent reverse flow when the injection pump was delivering fuel to the injector. A manual shutoff valve was also placed in the line. Oil flow was shut off during tests as it was discovered that enough lube oil was getting into the propane to cause ignition without the glow plug (albeit with considerable smoke). The oil flow was turned on during warm-up to minimize plunger and barrel wear.

The majority of the testing was carried out using the double injector designed by SNC (Figure 2-7). This combined two Stanadyne pencil injectors into a single body for mounting in the standard fuel injector opening in the CFR cylinder head. The two fuels remained completely separate in the injection system, there being no interconnection in the SNC double injector body. Each injector was supplied with the appropriate fuel by its respective pump as shown in Figure 2-8.



1 - Injection pump "Bosch" model PFR 1K80A 427/11
2 - 6 mm diameter barrel, "Bosch" part number 1418405002-800
3 - Mild steel insert
4 - Oil supply tube
5 - Fitting "Swagelock" male connector SS-100-2-2
6 - O-ring
7 - Copper gasket

Figure 2-6. Modified fuel injection pump for liquid propane delivery.

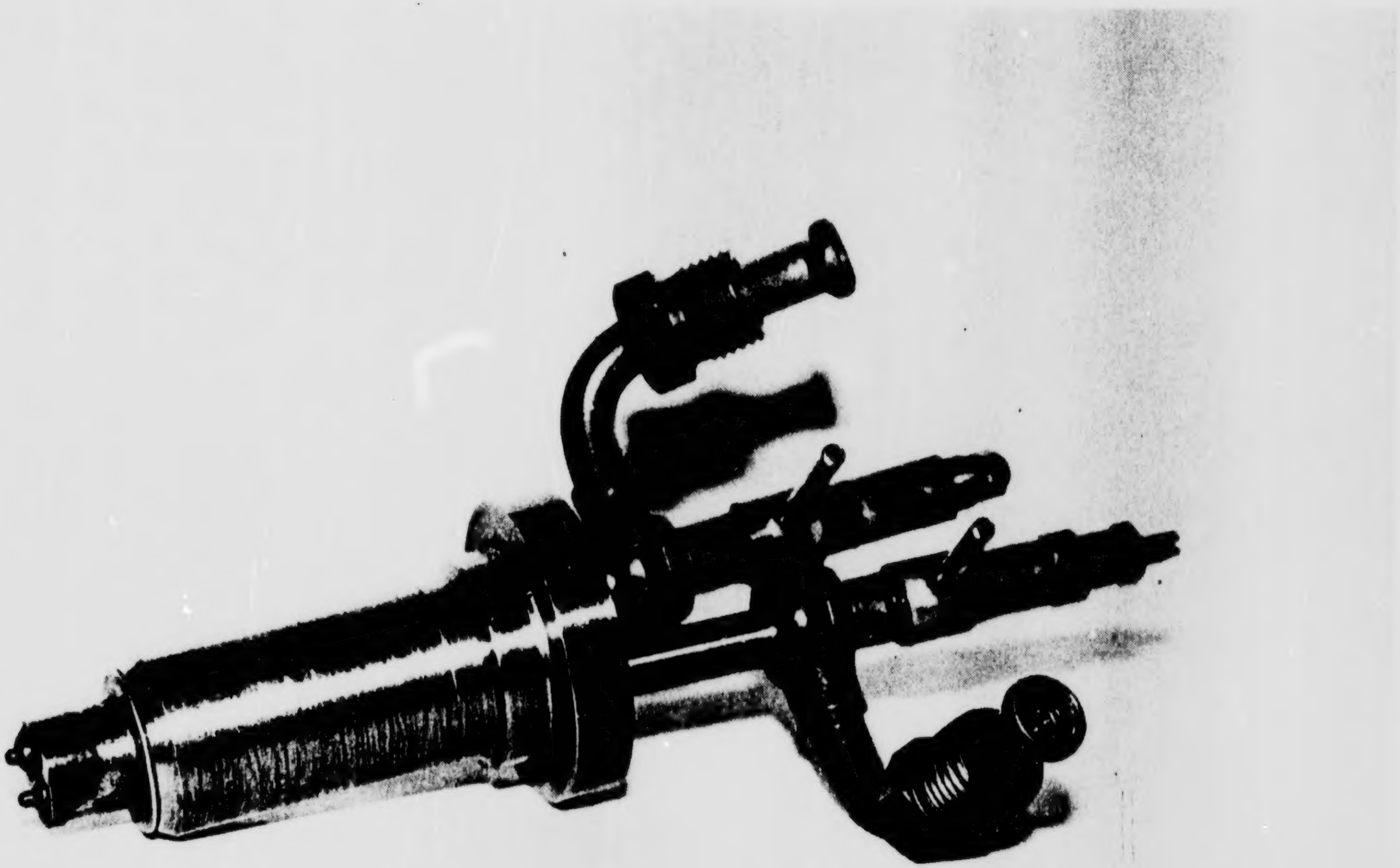


Figure 2-7. SNC Double Injector for Liquid Propane and Diesel Fuel with Modified Connections for Fuel Leakoff Return

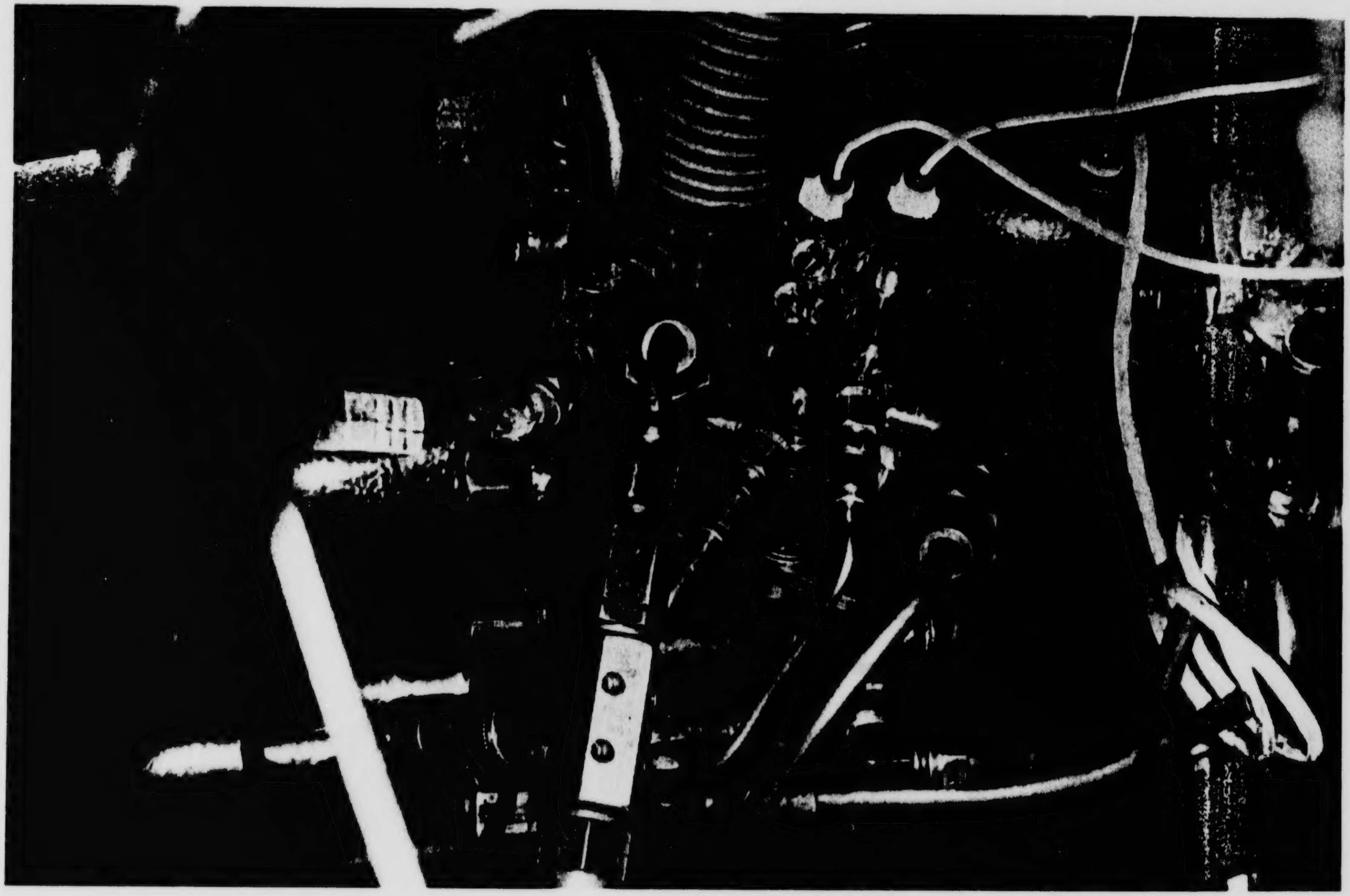


Figure 2-8. SNC Double Injector for Liquid Propane and Diesel Fuel Installed on the CFR Engine.

Some tests were carried out with two separate injectors. A Stanadyne Model 27333 injector (replacement for model 22202) was installed in the threaded access hole at the top of the combustion chamber and used to deliver propane (During tests of the SNC double injector, this access hole was fitted with a glow plug and adapter as shown in Figure 2-2, although the glow plug was never energized during the tests). A Stanadyne model 20502 injector was installed in the normal CFR injector location and used to deliver the diesel pilot spray to the combustion chamber. These injectors are shown in Figure 2-9.

2.2 TEST EQUIPMENT

The overall layout of test equipment and instrumentation is shown schematically in Figure 2-3 and pictorially in Figure 2-10. The following sections present the specifications for the test equipment and instrumentation used.

2.2.1 Dynamometer

"GENERAL ELECTRIC" D.C. Cradle-Type, Model TLC-6
8 HP (5.97 kW) 1400/4000 RPM as generator
5 HP (3.73 kW) 1250/4000 RPM as motor

2.2.2 Torque and Speed Measurement

2.2.2.1 Load cell: "INTERFACE" SUPER MINI type SM-100
Range: 100 lb (444.8 N)
Non-linearity: ±0.03% rated output
Hysteresis: ±0.02% rated output
Non-repeatability: ±0.01% rated output
Compensated temperature range: 0-150°F
Temperature effect on zero or rated output: ±0.08% rated output/100°F

2.2.2.2 Speed transducer: 60 teeth gear and "AIRPAX" 1-0002 magnetic pick-up

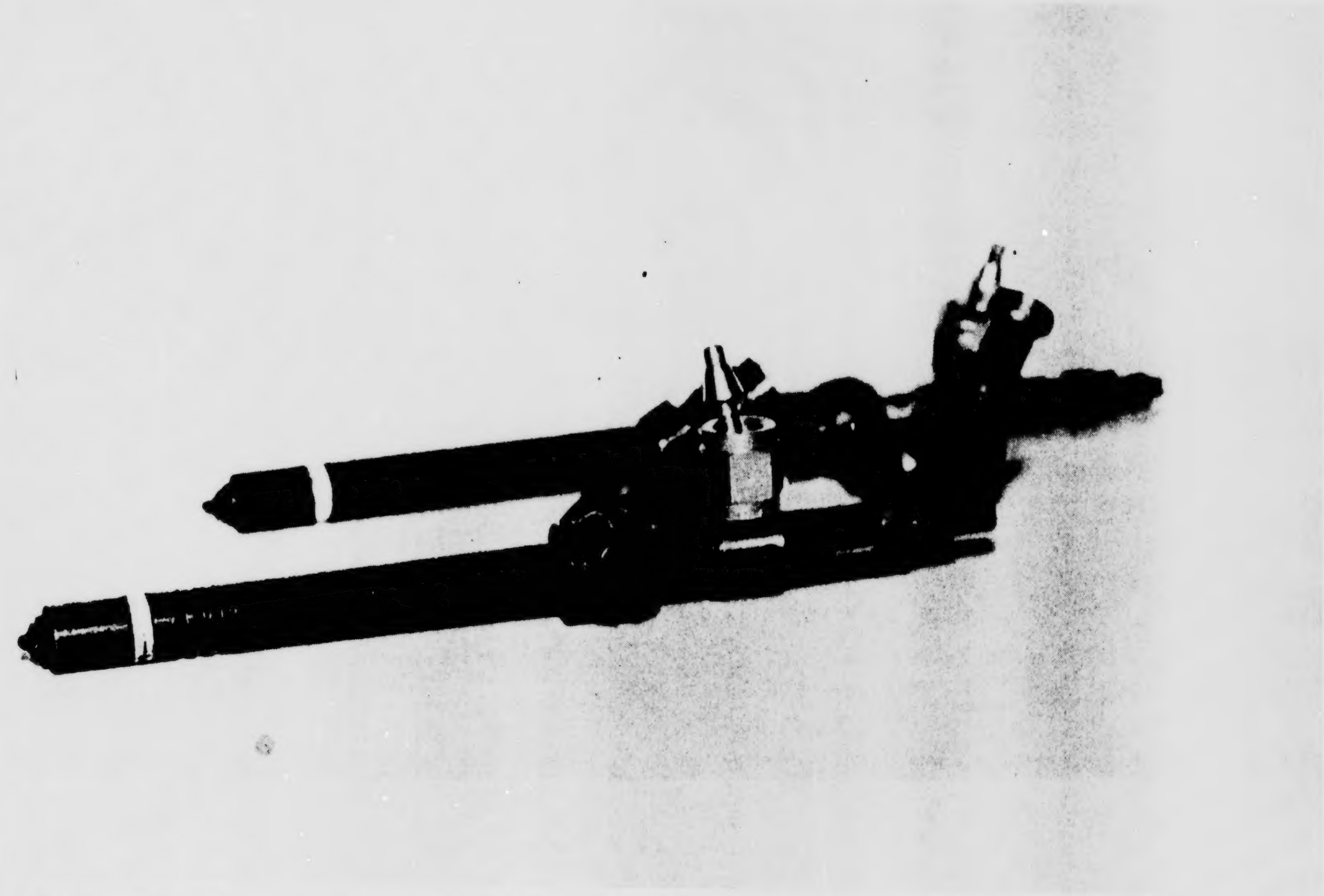


Figure 2-9. Separate Injectors Used for Liquid Propane and Diesel Fuel with Modified Connections for Fuel Leakoff Return.

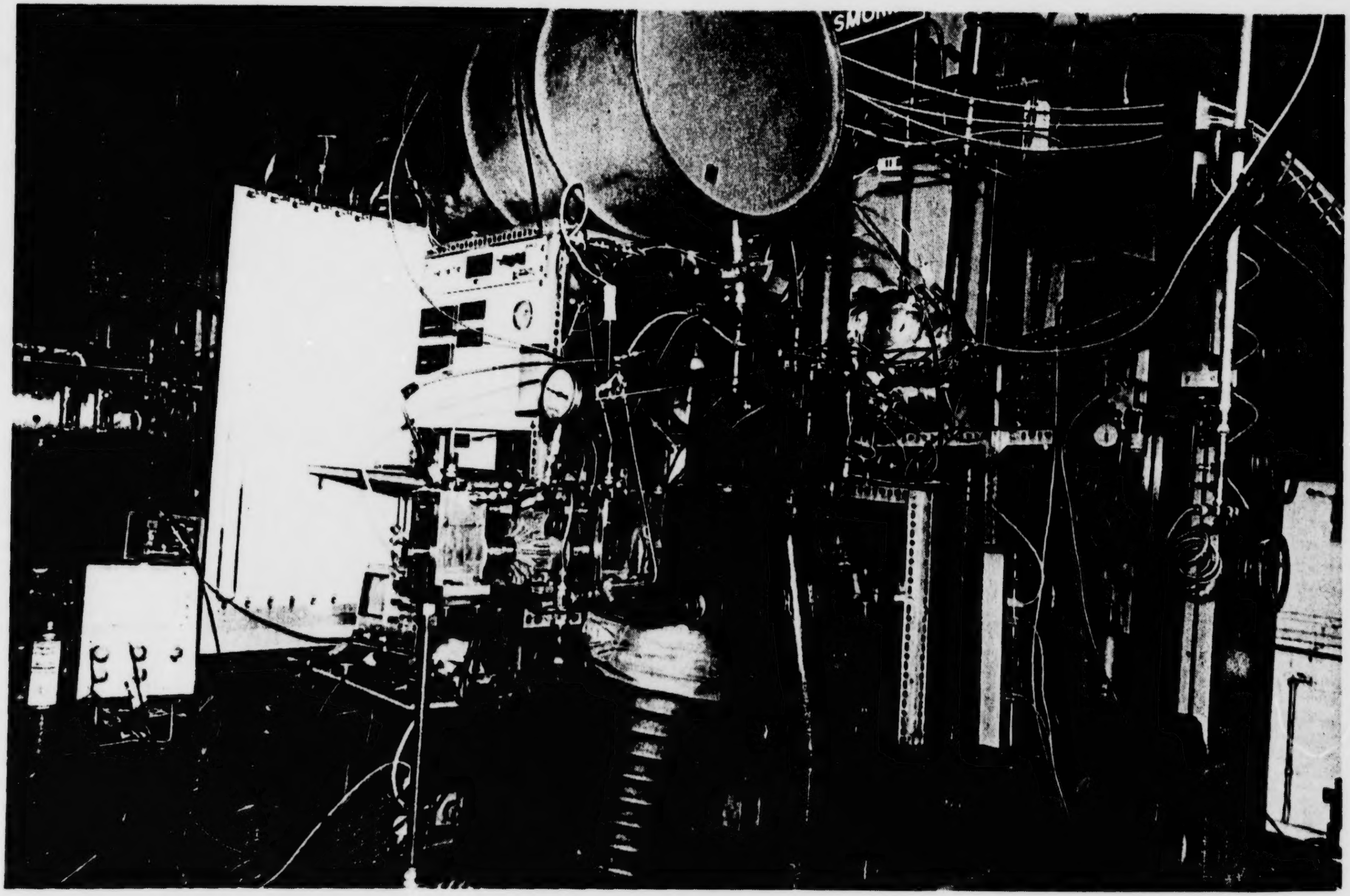


Figure 2-10. Overall View of the Experimental Apparatus and Instrumentation.

2.2.2.3 Indicators: "DAYTRONIC"
Torque - model 3378; excitation 3.28 kHz, regulated 2V
Speed - model 3340
Digital display resolution: 1 part in 5,000

2.2.3 Fuel Consumption Measurement

Fuel consumption measurements were made on a mass basis. Two fuel tanks were used for each fuel, a service tank and a weigh tank, the latter supported by an electronic balance. After recording the mass of the weigh tank, activation of an electronic timer switched fuel delivery from the service tank to the weigh tank for a preset interval. The final weight of the weigh tank could then be determined when fuel delivery was switched back to the service tank.

2.2.3.1 Propane consumption: Balance "METLER" model PK 36
Range: 0 to 30,000 g
Resolution: 0.1 g in 0 to 6,000 g range ("Delta Range")
1.0 g in 6,000 to 30,000 g range
Calibration: self, 16,000 g
Repeatability: <0.1 g in "Delta Range"
<0.5 g
Linearity: ±0.1 g in "Delta Range"
±1.0 g
Temperature range: 0 to 40°C

2.2.3.2 Diesel fuel consumption: Balance "SARTORIOUS" model 1404 MP8
Range: 2200 g
Resolution: 0.01 g
Linearity: ±0.015 g
Temperature range: 0 to 40°C

2.2.4 Smoke Measurement

"CELESCO" model 107 in-line exhaust smokemeter
Measuring method: attenuation of the intensity of a collimated light beam by smoke aerosol absorption and scattering

Scales: opacity or smoke density (K)
Opacity range: 0 to 100%
Resolution: 0.1%
Accuracy: ±0.5%
Smoke density range: 0 to 20 m^{-1}
Resolution: 0.01 m^{-1}
Accuracy: 0.1 m^{-1} for ≤ 2 m^{-1}
±5% of reading > 2 m^{-1}

2.2.5 Temperature Measurement

"DORIC" TRENDICATOR series 400A with 10 channels switching box and "OMEGA" thermocouples type K (Chromel-Alumel)

Range: -200 to 1,250°C
Accuracy of thermocouple: ±2.2°C or 0.75%, whichever is greater, for ≥ 0°C
Accuracy of indicator: ±0.5°C
Junction: grounded

2.2.6 Pressure Measurement

2.2.6.1 Cylinder pressure

- "AVL" type 12QP505 cl radiation shielded, water cooled quartz transducer

Measuring range: 0-150 bar dynamic load
Temperature range: ≤ 240°C
Sensitivity: in 34 pC/bar range, individually calibrated
Linearity: < ±1%
Temperature effect: 0.01% FS/°C
Natural Frequency: 65 kHz

- "AVL" model 3059B-A02.1 charge amplifier

Sensitivity ranges: 10^{-5} to 1 V/pC
Frequency response: near DC to 100 kHz (-3 dB)
Linearity error: <0.05%
Temperature drift: 0.2 to 2 mV/°C
Operating temperature range: 0 to 60°C

2.2.6.2 Fuel line pressure

- "AVL" type 31 DP strain-gauge transducer

Construction: full bridge, 120 ohms
Measuring range: 500 kg/cm^2
Non-linearity: <0.5%
Operating temperature range: -10 to 90°C
Temperature influence on zero or sensitivity: <0.04%/°C
Natural frequency: approximately 50 kHz

- "AVL" type 3009-A01 strain-gauge amplifier

Bridge supply voltage drift: <0.003%/°C
Bridge voltage accuracy: ±0.05%
Output frequency response: 0 to 70 kHz flat, unfiltered
Linearity: 0.03% FS
Zero drift: <1 µV/°C
Operating temperature: 0 to 50°C

2.2.6.3 Inlet air, exhaust, blowby pressures: U-tube mercury and water manometers

2.2.7 Injector Needle Lift Measurement

- "WOLFF"-ANALOG SENSOR integrated Hall effect transducer

Supply voltage: maximum 16 V DC
Magnetic flux density: no limit
Sensitivity: 1 V/mm
Frequency response: 25 kHz (-3 dB)
Thermal drift: -3 mV/kGauss-°C
Operating temperature: -40 to 150°C

- UTME (University of Toronto, Mechanical Engineering) instrumentation amplifier

For details of installation of the transducer on the STANADYNE injector, see Figure 2-11.

2.2.8 Air Rate of Flow

"TSI" model 2013 with air linearizer card and indicator model 67

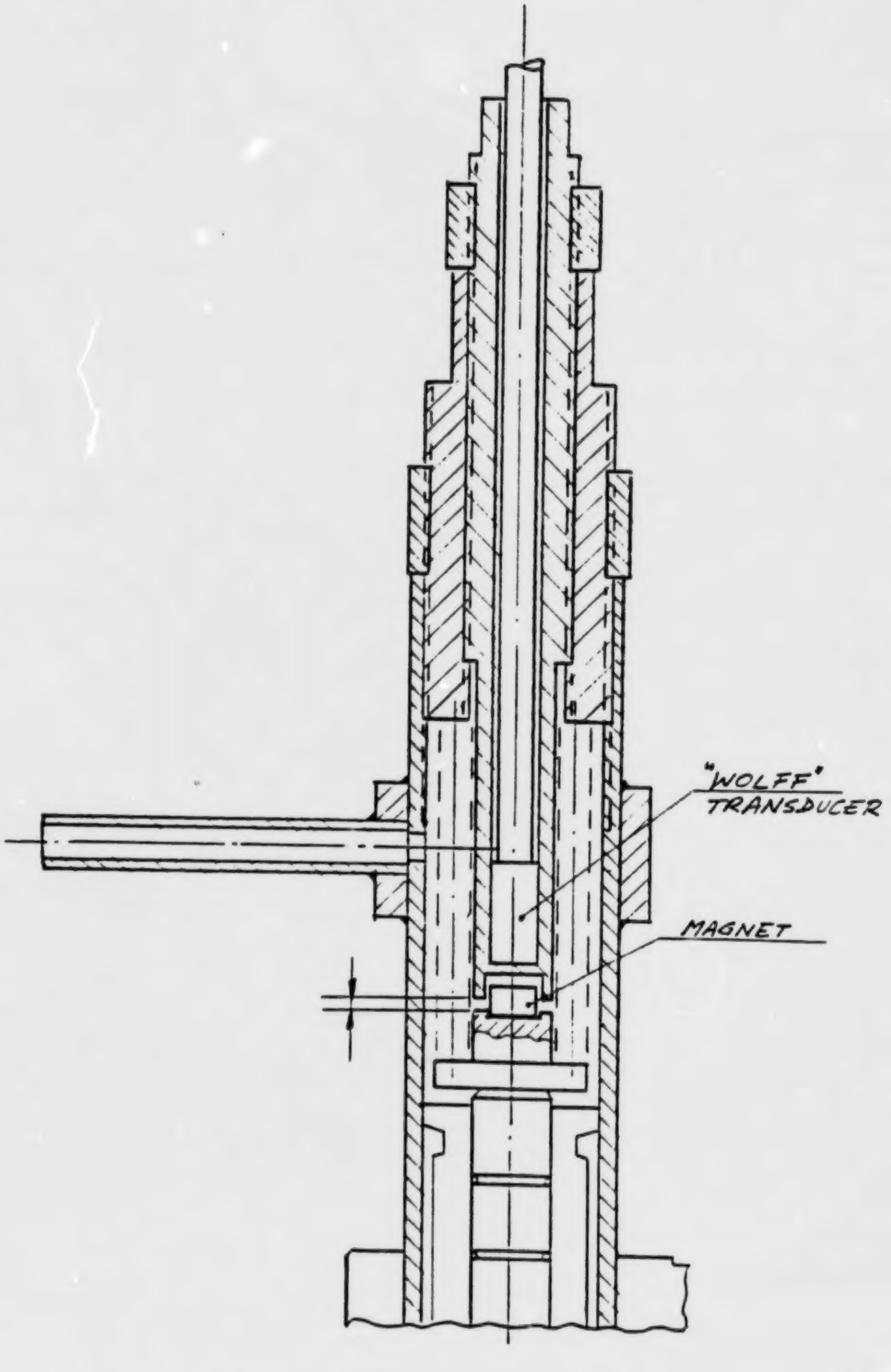

"WOLFF"
TRANSDUCER
MAGNET

Figure 2-11 Needle lift transducer installed on Stanadyne injector.

Accuracy: ±2% reading ±0.2% FS
Repeatability: ±0.1% reading ±0.05% FS
Response time: 0.1 s
Temperature compensation: 0 to 40°C
Operating pressure: 100 ± 20 kPa

2.2.9 Emissions

2.2.9.1 Hydrocarbons analyzer

- "BECKMAN" model 404

Measuring method: flame ionization method
Sample treatment: heated filter, supply line and analyzer, adjustable from 30 to 205°C (set at 191°C)
Ranges: 10, 25, 100, 250 ppm, 0.1, 0.25, 1.0% hydrocarbons as methane
Response time: <1.5 s detection module
5.5 s complete system (48" sample line) for 95% of final reading

2.2.9.2 NO analyzer

- "BECKMAN" model 951

Measuring method: chemiluminiscence
Ranges: 10, 25, 100, 250, 1000, 2500, 10,000 ppm
Sensitivity: 0.1 ppm on 10 ppm range
Linearity: ±1% FS
Precision: 0.5% FS
Dynamic response: ≤1 s

2.2.9.3 O_2 analyzer

- "BECKMAN" model OM-11

Measuring method: polarographic
Range: 25%
Accuracy: ±1% FS
Linearity: ±1.2% O_2
Response speed: 90% within 300 to 600 ms

2.2.9.4 CO and CO_2 analyzers

- "BECKMAN" model 864

Measuring method: non-dispersive infrared (NDIR)
Ranges: 2, 10% CO
20% CO_2
Precision: 1% FS
Noise: 1% FS
Zero or span drift: ±1% FS for 24 h
Response speed: adjustable, 90% in 0.5 to 26 s

2.2.9.5 Exhaust sample conditioning system for NO, O_2, CO and CO_2 analyzers

- "BECKMAN", with diaphragm compressor and air dryer.

2.2.10 Crank Angle Encoder

"AVL" model 360 C/600/S optical infrared encoder
Trigger pulse: 1 low pulse/revolution, can be aligned with top dead centre or other required crank shaft event
Crack degree marks: maximum 3,600 signal edges/revolution (0.1 crank angle resolution)
Speed range: 250 to 8,000 rpm
Temperature range: 0 to 60°C
Accuracy: ±0.05° crank angle

2.2.11 Dynamic Display and Measurement (for cylinder and fuel line pressures, injector needle lift and crank angle marks)

2.2.11.1 Four channel storage oscilloscope

- "TEKTRONIX" model 5103N system

Calibrated deflection: 1 mV to 5V/division
Band width: DC to 1 MHz (usable)
Accuracy: within 2%
Time base: 1 μs to 5 s/division
Time base accuracy: within 3 to 5%

2.2.11.2 Two channel oscilloscope

- "B+K PRECISION" model 1476

Calibrated deflection: 10 mV to 20 V/division
Band width: DC to 10 MHz (-3 dB)
Accuracy: ±5%
Time base: 1 μs to 0.5 s/division
Time base accuracy: ±5%

2.2.12 dP/dt Adapter

UTME construction. Supplied with the output of the cylinder pressure amplifier, the output is the differential of this signal.

3. TEST PROCEDURE

A layout of the test equipment and instrumentation is presented in Figure 2-3.

During the test, the following parameters are recorded:

- engine speed - rpm
- engine torque - N.m
- engine friction torque - N.m (at stabilized working temperature, using the dynamometer motoring capability)
- mass fuel consumption - g
- test duration - s
- inlet air flow - standard L/min.
- inlet pressure drop - in.Hg
- exhaust back pressure - in.Hg
- blow-by differential pressure - in.Hg (to determine flow)
- inlet air temperature - deg.C
- exhaust temperature at engine - deg.C
- oil temperature - deg.C
- water temperature - deg.C
- smoke opacity - %
- HC emission as CH_4 - ppm
- NO/NO_x emission - ppm
- O_2 emission - %
- CO emission - %
- CO_2 emission - %
- cylinder pressure - bar (x 100 = kPa)
- dP/dt - bar/deg
- oil pressure - kPa
- fuel line pressure - bar
- beginning of injection - deg BTDC
- end of injection - deg ATDC
- maximum needle lift - mm
- combustion start - deg BTDC
- atmospheric pressure - kPa
- atmospheric vapour pressure - kPa

From those values, a number of derived parameters are determined through computation:

- ignition delay - deg
- standard atmospheric conditions correction factor
- fuel-air ratio
- corrected indicated power - kW
- corrected brake power - kW
- indicated specific fuel consumption - MJ/KW.h
- brake specific fuel consumption - MJ/kW.h
- indicated mean effective pressure - bar
- brake mean effective pressure - bar
- indicated efficiency
- brake efficiency

After setting of desired compression ratio and calibration of instrumentation, the engine-dynamometer system is maintained at the desired speed by control of dynamometer load and fuel rack position.

All data are recorded under stabilized thermal conditions.

4. RESULTS AND DISCUSSION

In this section test results for three methods of operating the engine will be compared. These are the original diesel fuel injection, propane fumigation with diesel pilot injection and dual fuel operation injecting both propane and a diesel pilot through separate fuel injection systems. The data presented in all cases was with the injection timing resulting in best power output, limited to a maximum rate of pressure rise of 345 kPa/degree of crankshaft rotation (50 psi/degree). Higher rates of pressure rise resulted in excessive engine vibration and noise.

For the dual fuel operation, both fuel injectors were combined into a single body which was mounted in the location provided for the original equipment diesel fuel injector. This double injector was provided by SNC. Initial tests were conducted to determine the importance of the relative timing between the diesel and propane injections. It was found that over a range of about 5 degrees on either side of zero relative angle between the two injections, there was little effect on engine performance. Therefore, all subsequent dual-fuel tests were conducted with no phase shift between the two injection pumps. In all dual-fuel tests, the minimum amount of diesel pilot injection to ensure stable operation was used. The percentage of the total fuel energy supplied by the diesel fuel therefore varies from test to test. This data will be presented subsequently.

The power output of diesel engines is normally limited by the level of smoke emissions that can be tolerated. Figures 4-1 through 4-9 show the smoke emissions of the test engine as a function of engine load (expressed as indicated mean effective pressure, imep) for the three compression ratios and three engine speeds tested. It may be seen that both fumigation and dual-fuel operation significantly reduce the smoke emissions, particularly at higher loads. Thus, for a given smoke level, an engine employing propane fumigation or dual-fuel operation can produce a greater power output. This is an important consideration as emission standards for heavy duty vehicles are about to be implemented.

Figures 4-10 through 4-18 show the percentage of total fuel mass that was diesel fuel ("diesel injection rate") as a function of engine load. Except at light loads with 19:1 CR, less diesel fuel is required for operation with dual-fuel injection. Increasing the compression ratio decreases the amount of diesel fuel required for ignition with the dual-fuel injection system. For all of the load and speed conditions tested, diesel fuel was required to provide about 10-15% of the total fuel energy, with the injected propane providing the rest. With fumigation, more diesel fuel is required as the compression ratio increases, but this is due to a knock limitation (maximum rate of pressure rise of 345 kPa/degree or 50 psi/degree) rather than failure to achieve ignition. Engine speed has little effect on the amount of diesel fuel required for either method of fueling except for fumigation at 25:1 CR where more propane can be tolerated at higher speeds for the same maximum rate of pressure rise. From a fuel substitution standpoint, dual-fueling is more desirable than fumigation because it permits a larger percentage of propane use.

Figures 4-19 through 4-27 show indicated specific fuel consumption, expressed in energy units, as a function of load for the three fueling methods. With the exception of the tests at 1200 rpm for the 22:1 compression ratio, the energy consumption of the dual-fuel injection engine was nearly the same as that of the original diesel. In general, the energy consumption is low at light loads, increasing at high loads as smoke begins to form. The reasons for the anomalous results at 1200 rpm with the 22:1 compression ratio are not known.

The fumigated engine has poorer fuel consumption at partial loads. The fuel consumption falls with increasing load until the fuel consumption of the propane fumigated engine is equivalent to that of the original diesel at full load. The poor part load efficiency is due to incomplete combustion. This may be seen in the plots of both hydrocarbon emissions (Figures 4-23 through 4-

36) and CO emissions (Figures 4-37 through 4-45), where the emissions of the fumigated engine are high at light loads and drop steadily as full load operation is reached. Hydrocarbon and CO emissions of the dual-fueled engine are low and comparable to those of the original diesel.

NO_x emissions are generally lower with dual fueling than with diesel fuel. Figures 4-46 through 4-54 show exhaust NO_x concentrations as a function of engine load. Increasing compression ratio lowers the NO_x emissions of the diesel-fueled engine more than those of the dual-fueled engine so that at 25:1 compression ratio the NO_x emissions are about the same from the two engines. The fumigated engine shows different characteristics since a mostly premixed charge is burning. The NO_x emissions are low at light loads when the propane air mixture is lean and increase sharply at the load is increased. This is due to the higher peak combustion temperatures reached in premixed propane-air charge as the mixture is richened to increase the engine load.

Additional tests were conducted at 19:1 compression ratio using two separate injectors rather than the single body injector provided by SNC. A Stanadyne model 27333 injector (replacement for model 22202) was installed in the threaded access hole at the top of the combustion chamber and used to deliver propane to the combustion chamber. The tip of this injector has four 0.011 in. diameter holes. A Stanadyne model 20502 injector was installed in the normal CFR fuel injector locaton and used to deliver the diesel pilot spray to the combustion chamber. The tip of this injector has five 0.007 in. diameter holes. None of these holes were plugged as was the case with the SNC double injector which had four of the five holes plugged.

The main difference in the results obtained from tests of the two injector configurations is in the fuel consumption. Figure 4-55 shows the indicated specfic fuel consumption as a function of engine load. At every speed, the engine had significantly lower fuel consumption with the SNC double injector than with the two separate injectors. The largest difference occurs at the lowest engine speed (1200 rpm) and is reduced as the speed increases. Since the swirl velocity in the prechamber increases with engine speed, this suggests that the swirl motion in the prechamber is inadequate to properly mix the pilot spray and the propane spray when the injectors are at opposite ends of the swirl chamber as was the case with the separate injectors. The diesel pilot spray becomes many individual ignition sources. With the double injector, the two injectors spray over much the same arc; thus, the ignition sources are well distributed throughout the propane and burning can take place rapidly. With the separate injectors, however, the vaporizing droplets of the diesel spray must be carried into the propane containng region by fluid motion. This results in a longer burn time and subsequently lower efficiency.

This hypothesis is supported by the significantly higher hydrocarbon emissions at light loads with the separate injectors as shown in Figure 4-56. As the propane is also mixing with air while the diesel fuel droplets are mixing in, some regions may be too lean to burn by the time an ignition source or flame front reaches them. The CO emissions (Figure 4-57) are also higher with the separate injectors than with the SNC double injector at the same speed. Figure 4-58 shows the smoke emissions as a function of engine load for the two injector configurations. Except for the highest speed tested (1800 rpm), the SNC double injector had significantly lower smoke emissions than the two separate injectors. Within the scatter of the experimental data, the NO_x emissions (Figure 4-59) were comparable for both injector configuations, increasing with increasing engine load.

Figure 5.60 shows that, except at 1500 rpm, the engine required more diesel fuel to achieve reliable ignition at light loads with the SNC double injector than with the two separate injectors. The reason for the anomalous behavior at 1500 rpm is not clear.

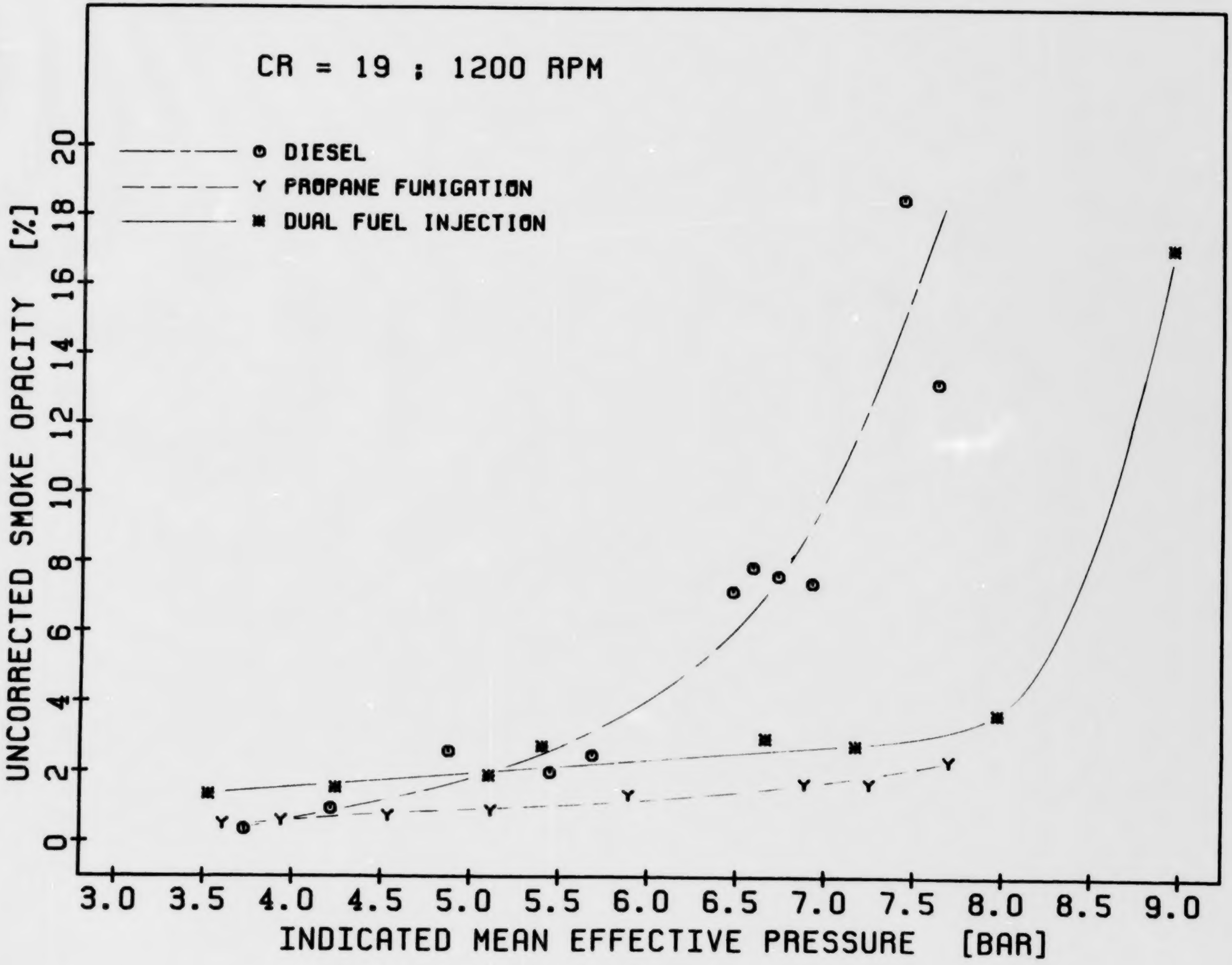

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
0
2
4
6
8
10
12
14
16
18
20
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-1

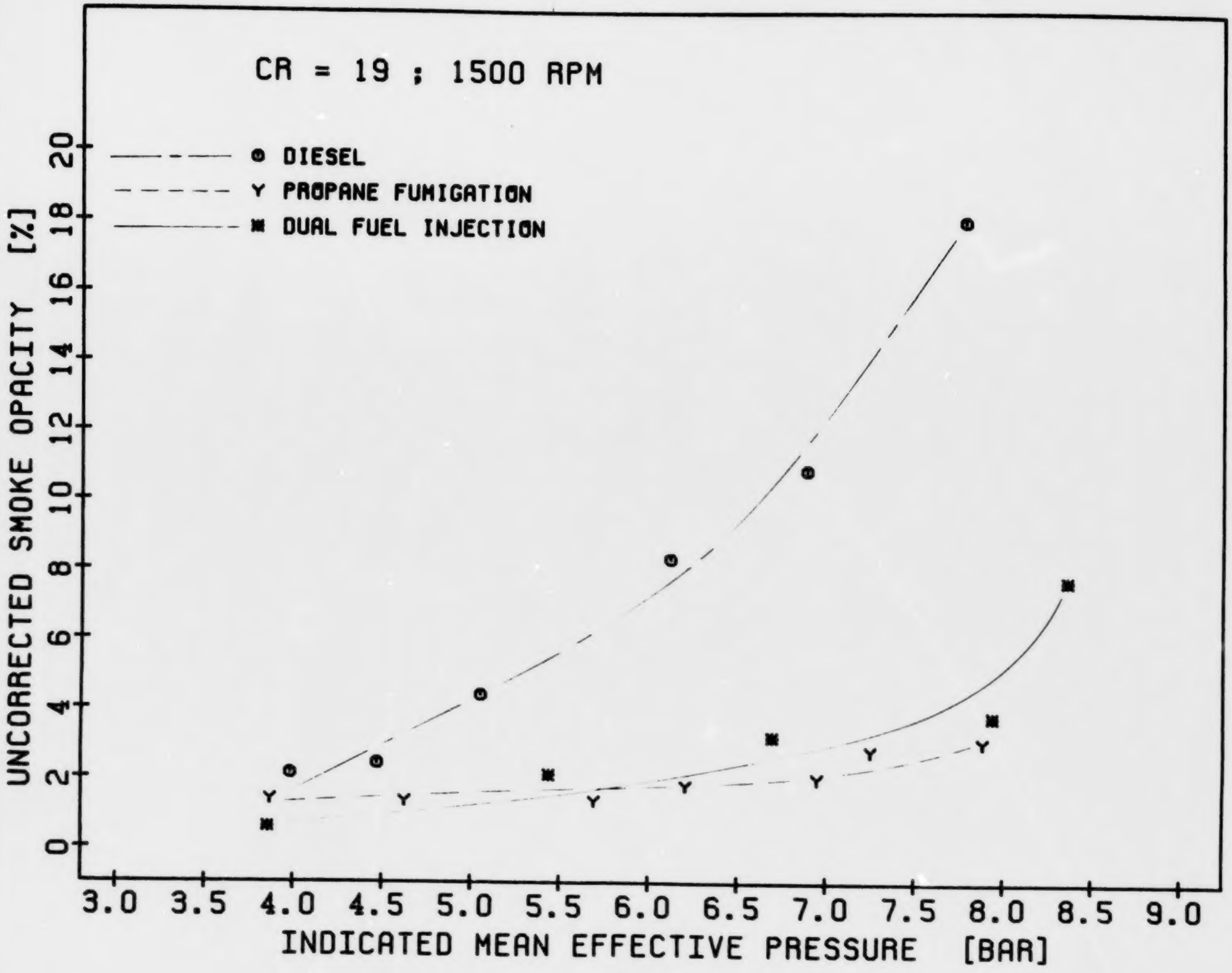

CR = 19 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
0
2
4
6
8
10
12
14
16
18
20
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-2

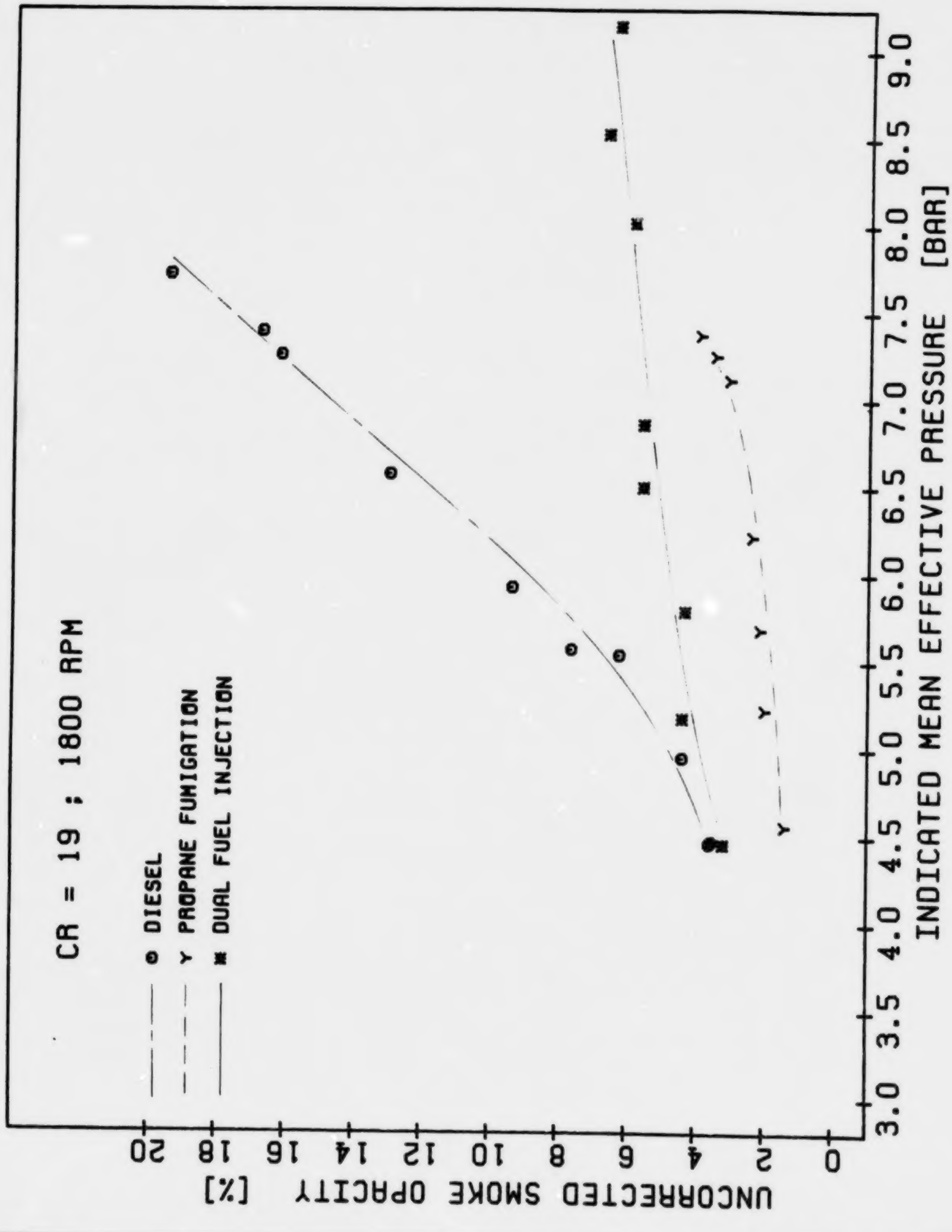

CR = 19 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
0
2
4
6
8
10
12
14
16
18
20
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-3

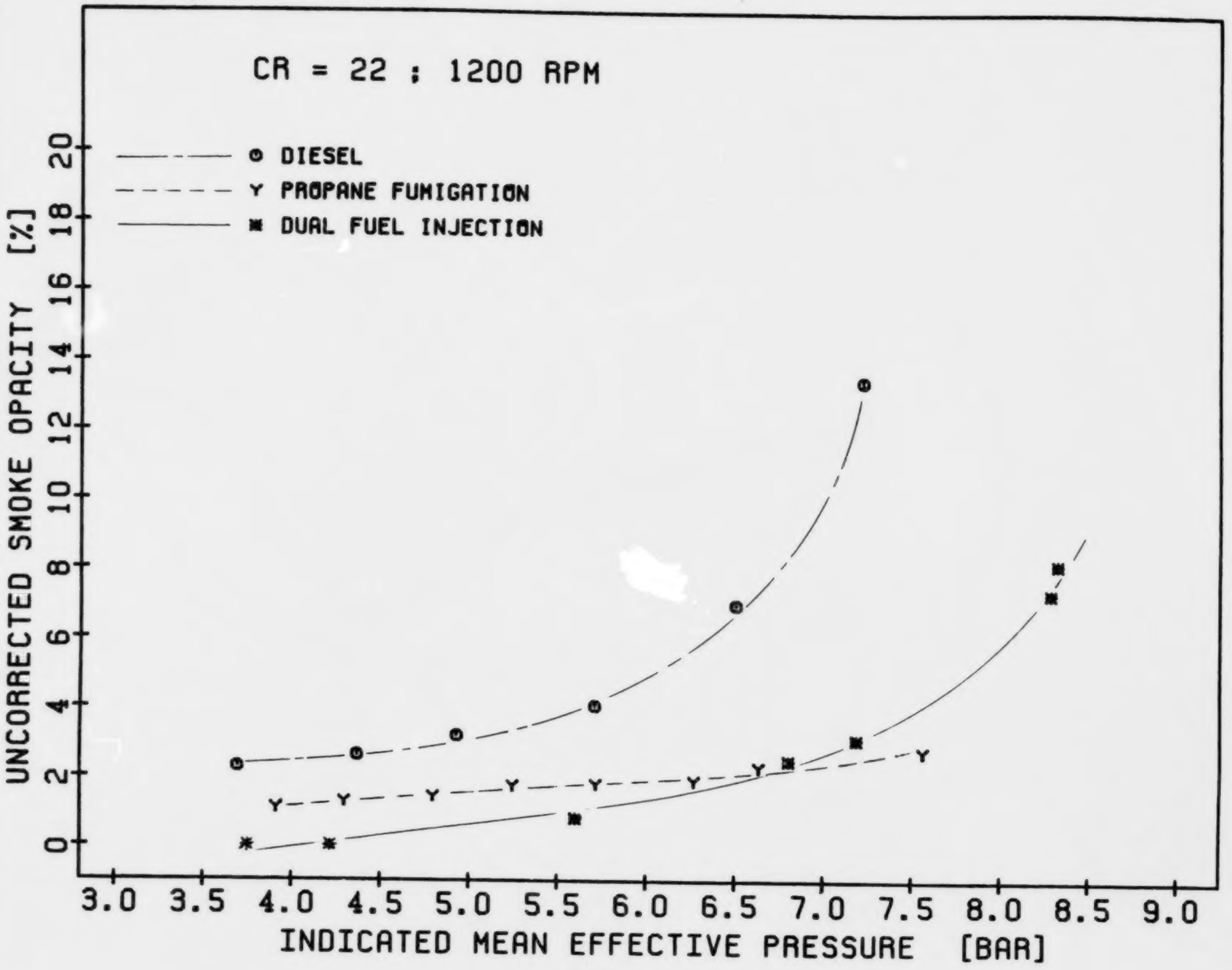

CR = 22 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
0
2
4
6
8
10
12
14
16
18
20
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-4

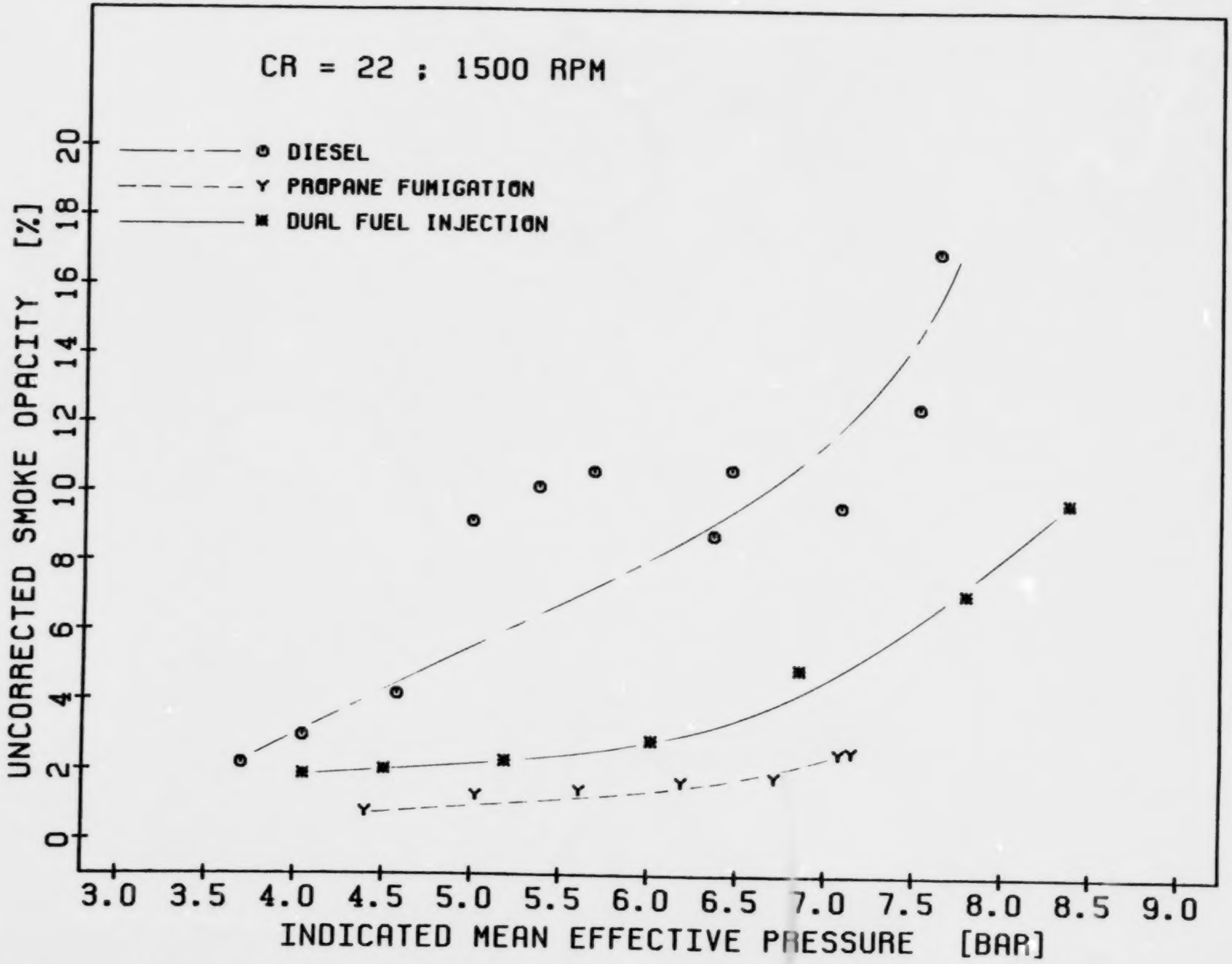
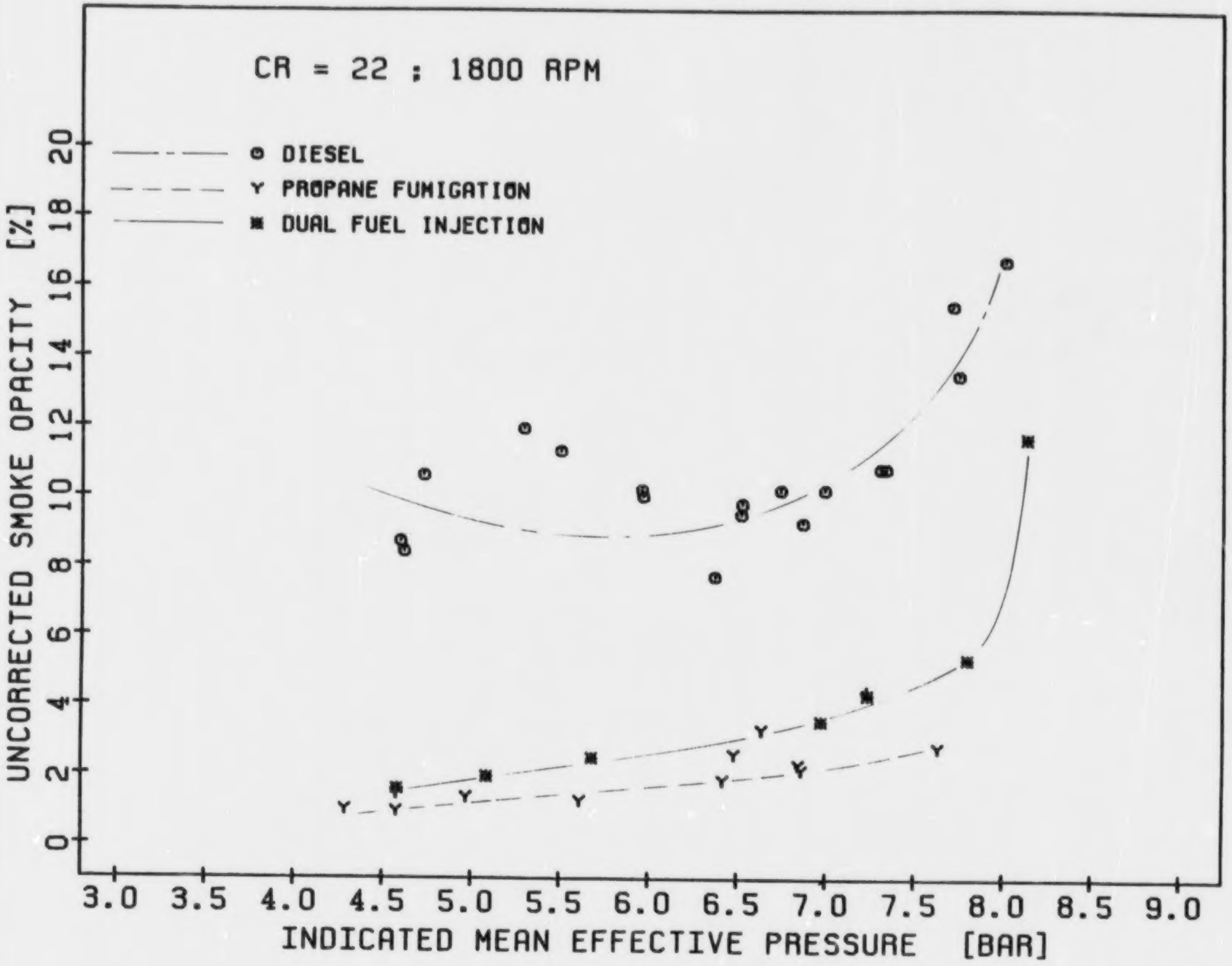

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
0
2
4
6
8
10
12
14
16
18
20
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-5


CR = 22 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
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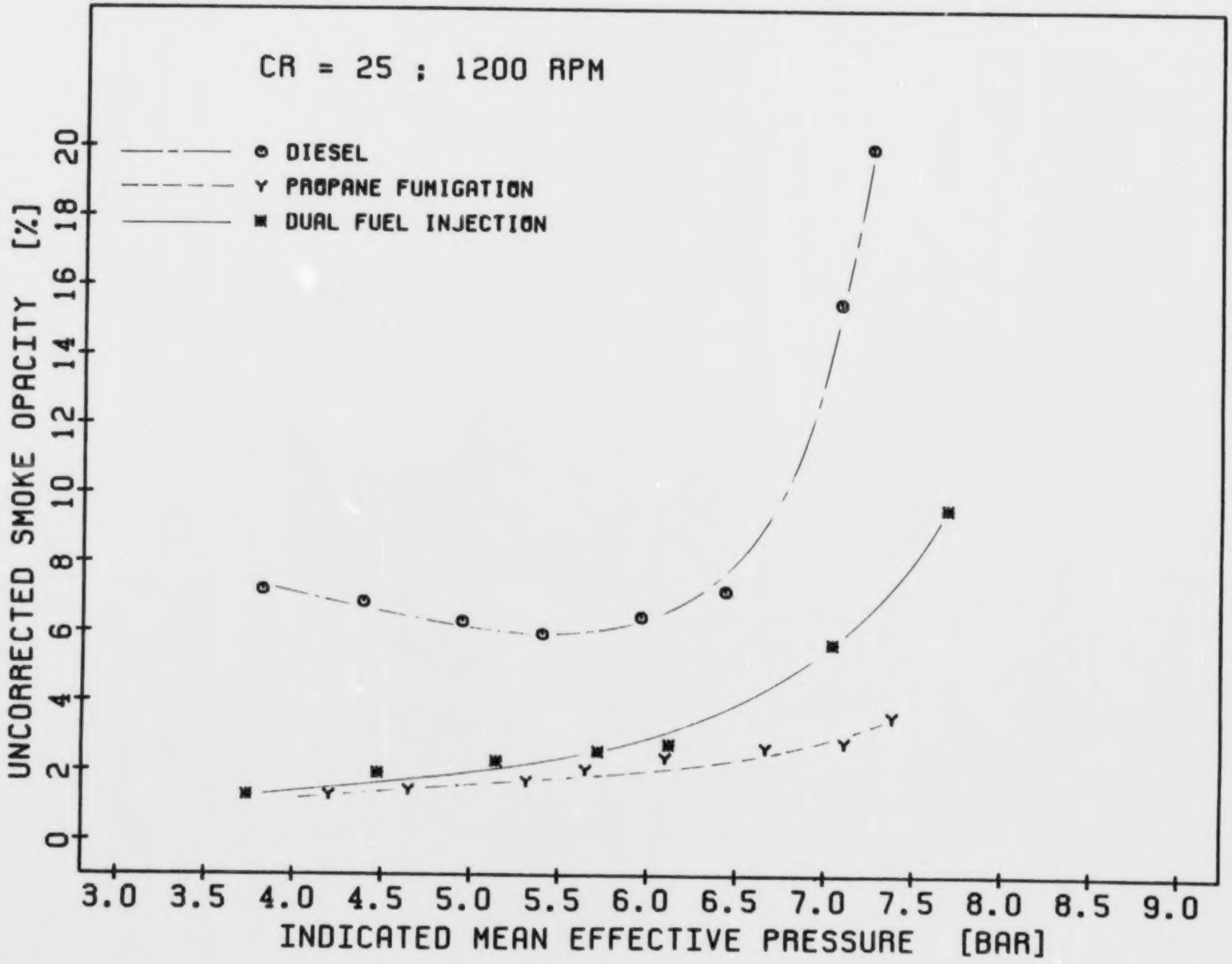
2
4
6
8
10
12
14
16
18
20
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-6


CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
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2
4
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12
14
16
18
20
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-7

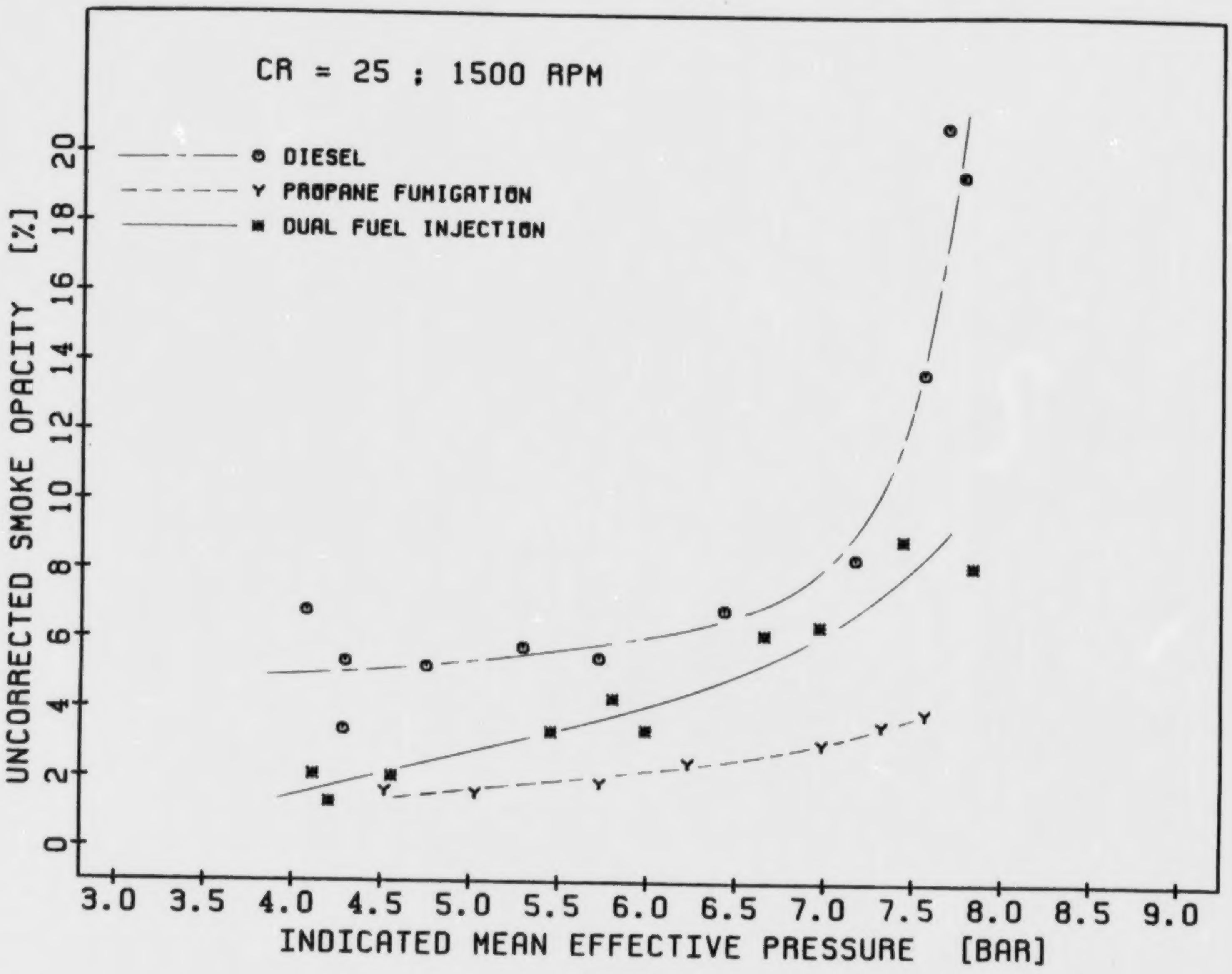

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
0
2
4
6
8
10
12
14
16
18
20
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-8

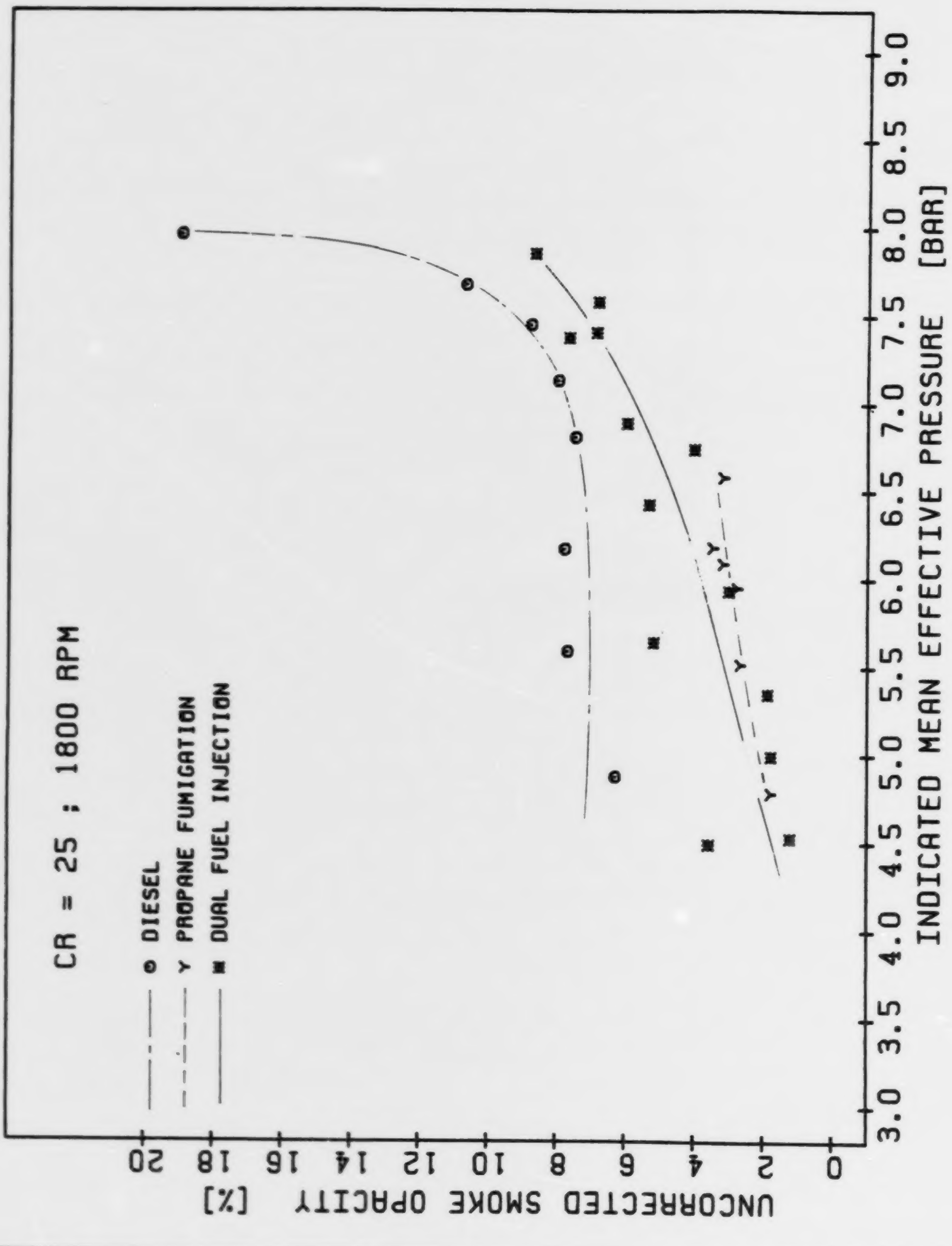

CR = 25 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
UNCORRECTED SMOKE OPACITY [%]
0
2
4
6
8
10
12
14
16
18
20
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-9

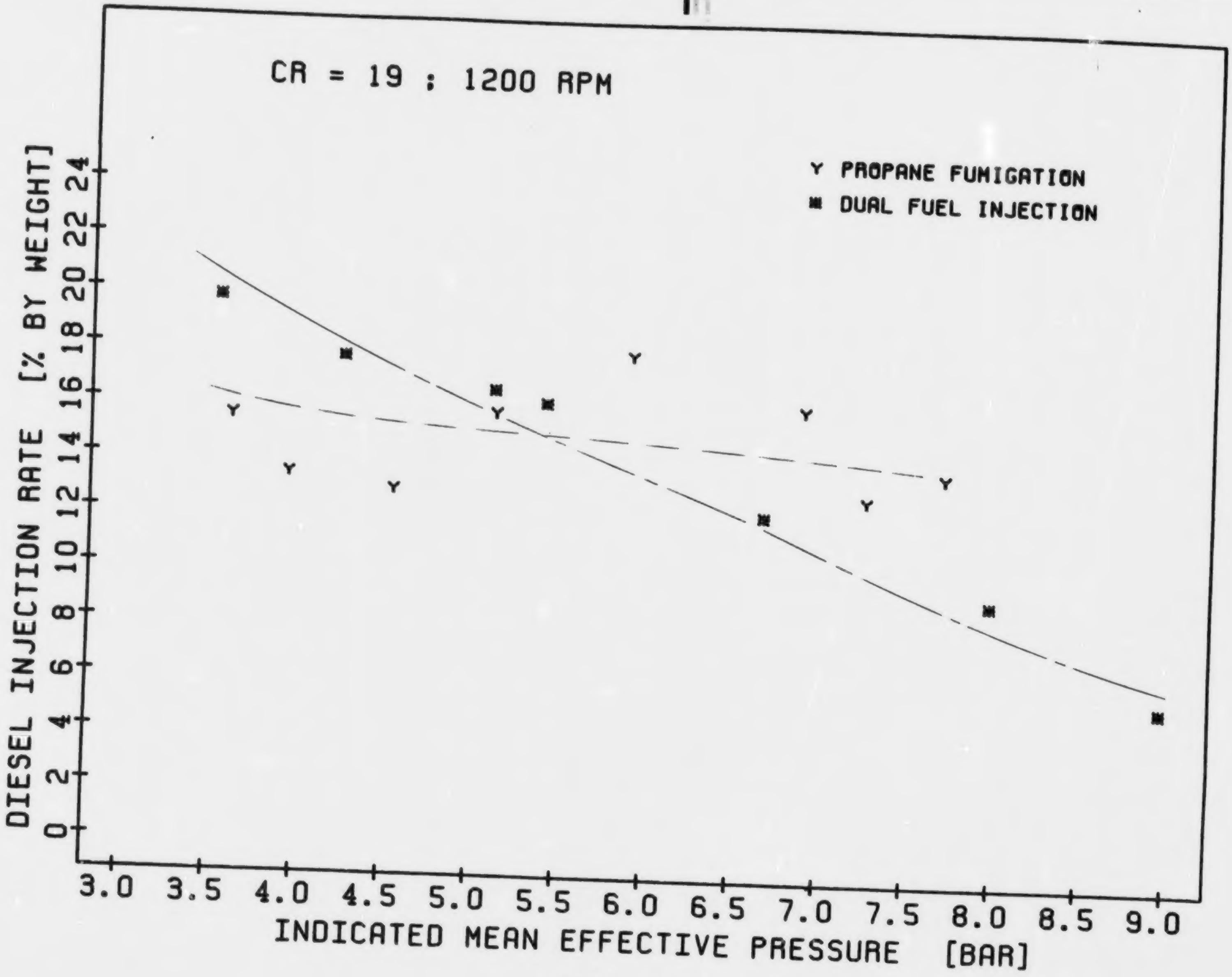

CR = 19 : 1200 RPM
Y PROPANE FUMIGATION
* DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-10

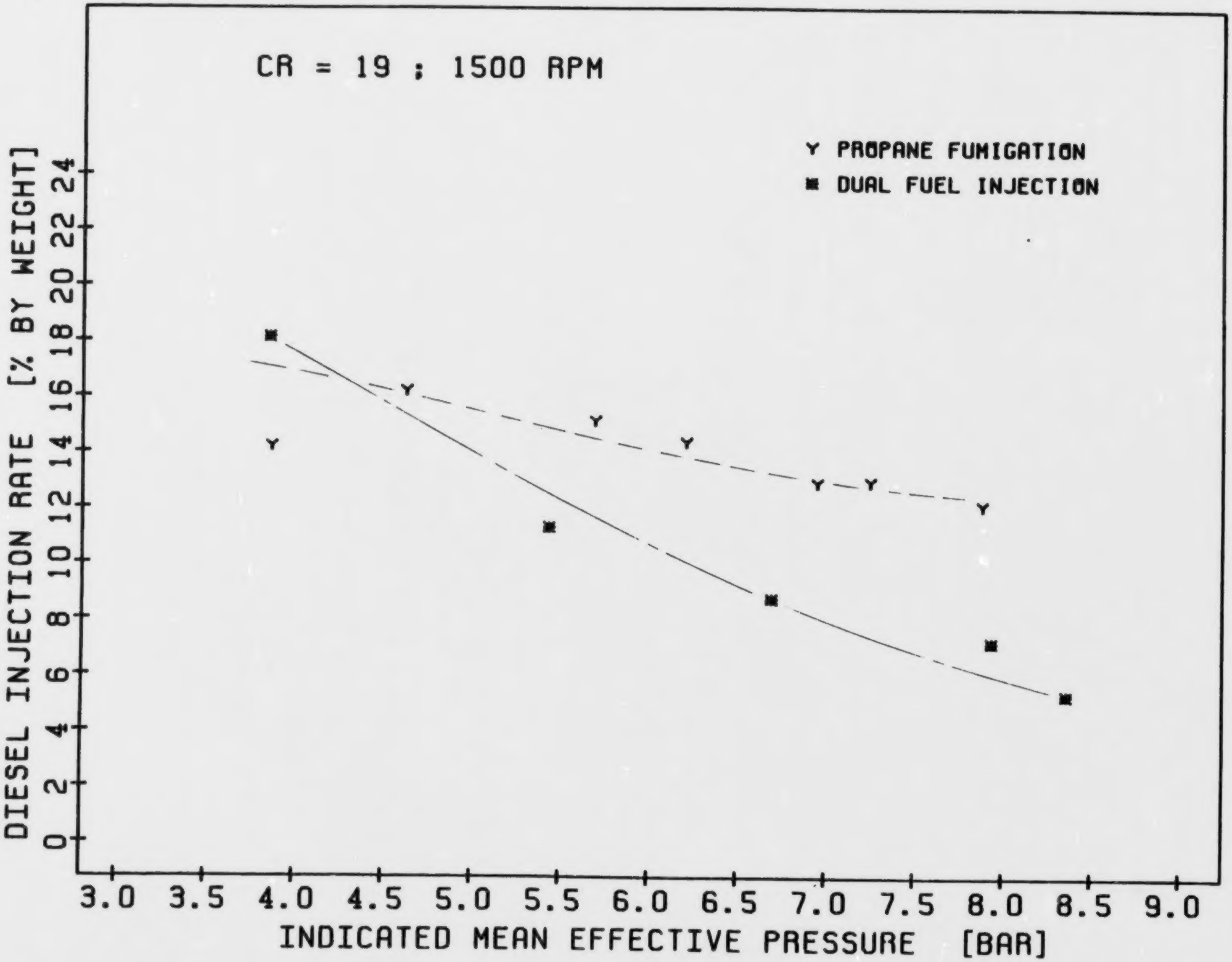

CR = 19 ; 1500 RPM
Y PROPANE FUMIGATION
■ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-11

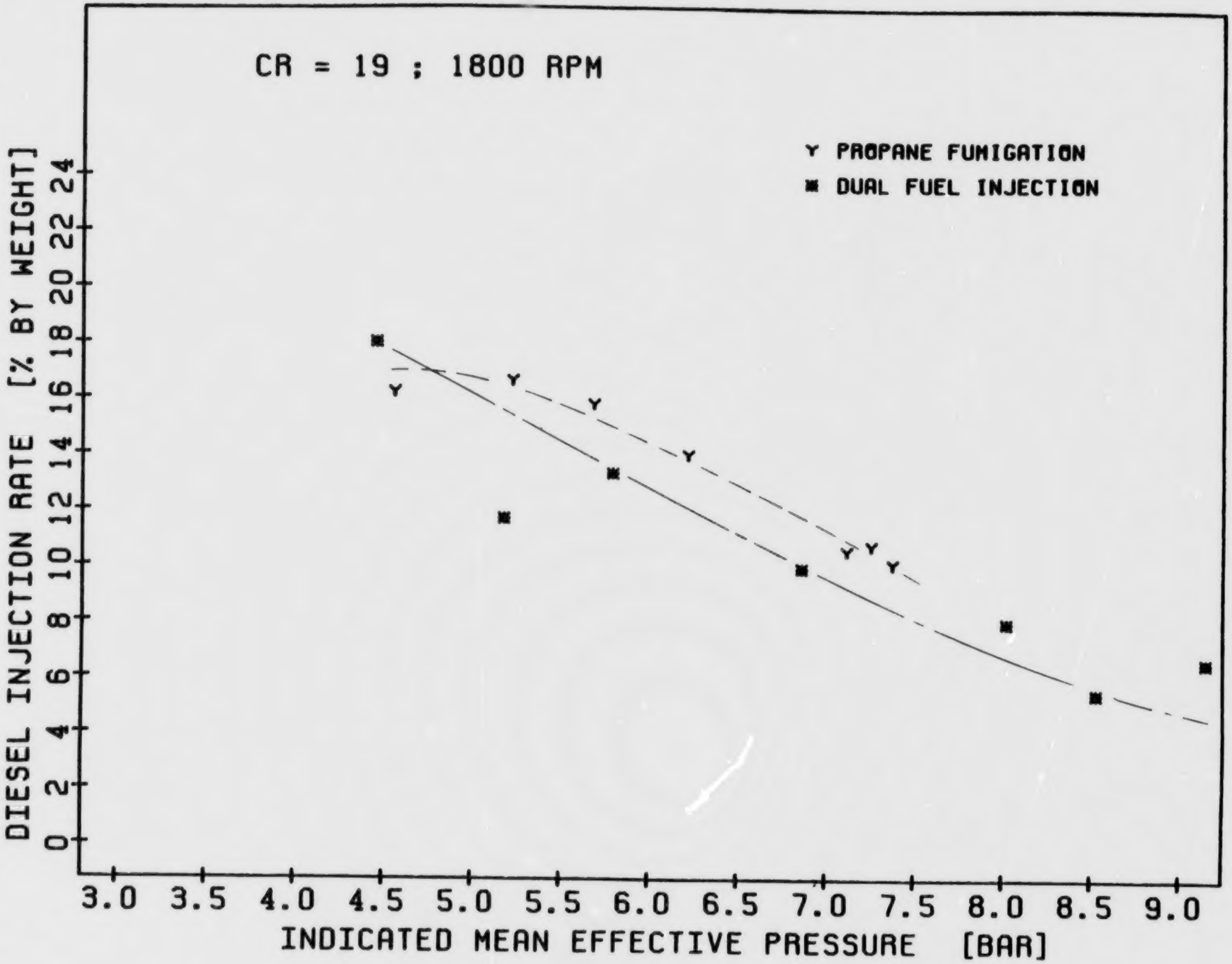

CR = 19 ; 1800 RPM
Y PROPANE FUMIGATION
■ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-12

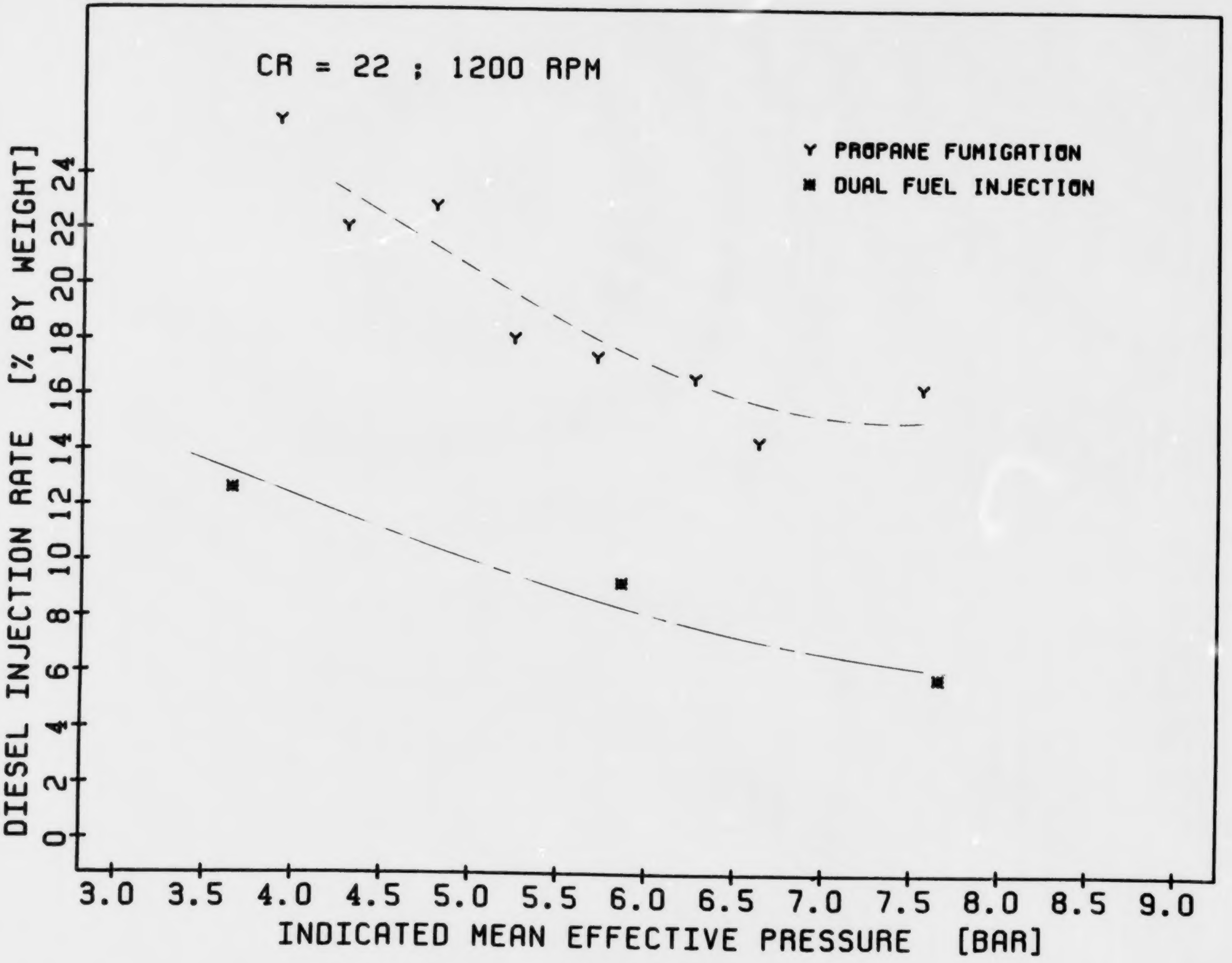

CR = 22 ; 1200 RPM
Y PROPANE FUMIGATION
✱ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-13

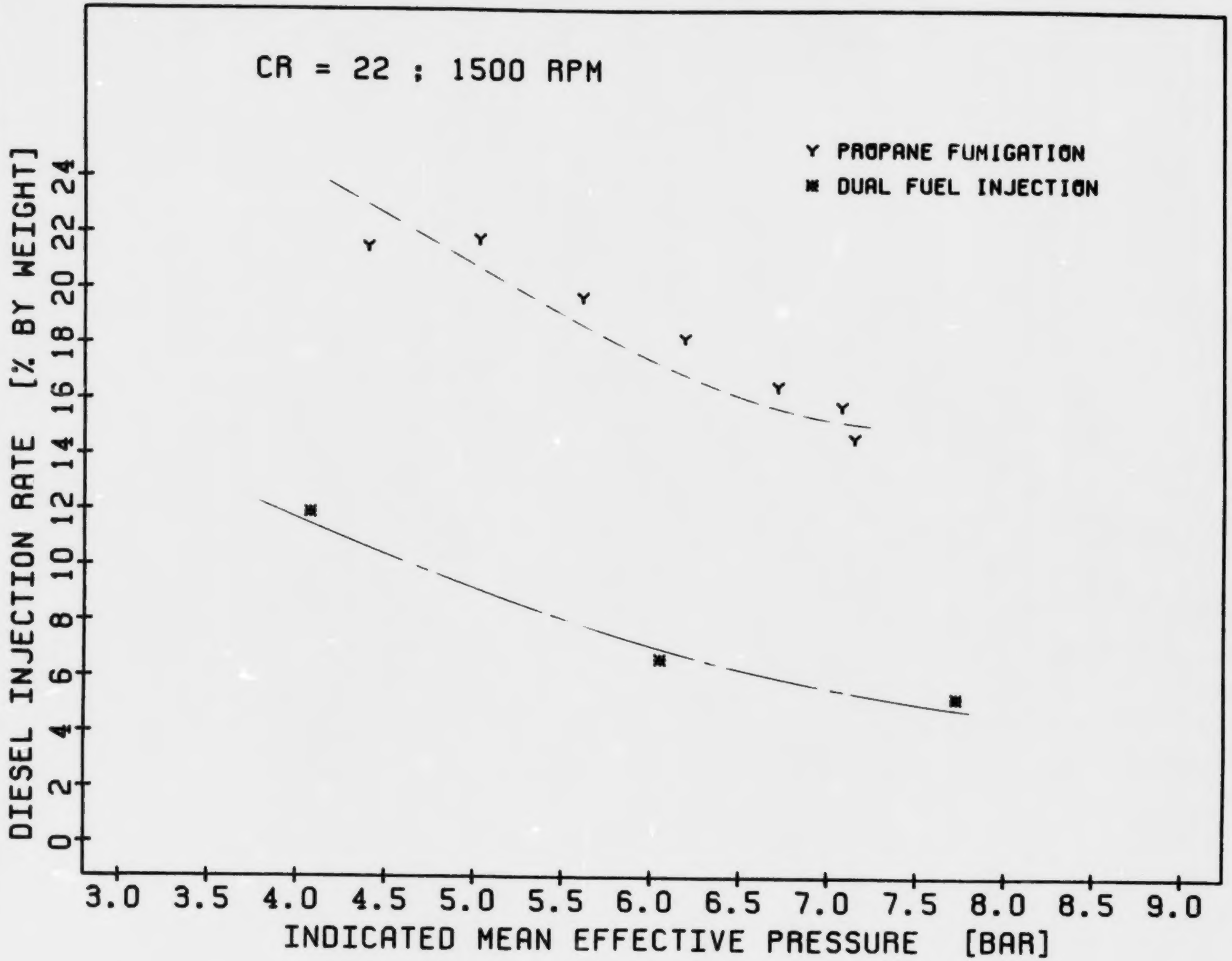

CR = 22 ; 1500 RPM
Y PROPANE FUMIGATION
✱ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-14

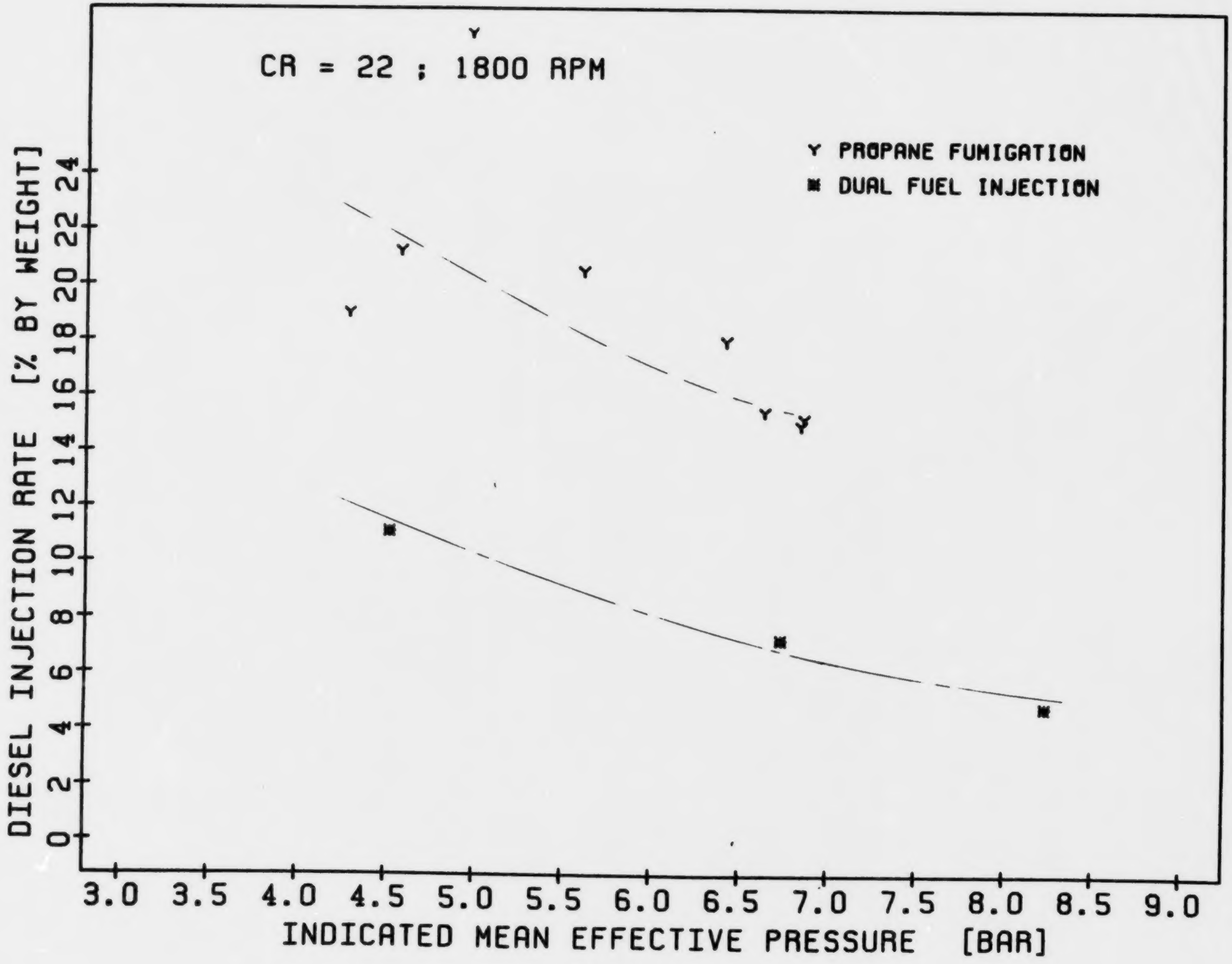

CR = 22 ; 1800 RPM
Y PROPANE FUMIGATION
✱ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-15

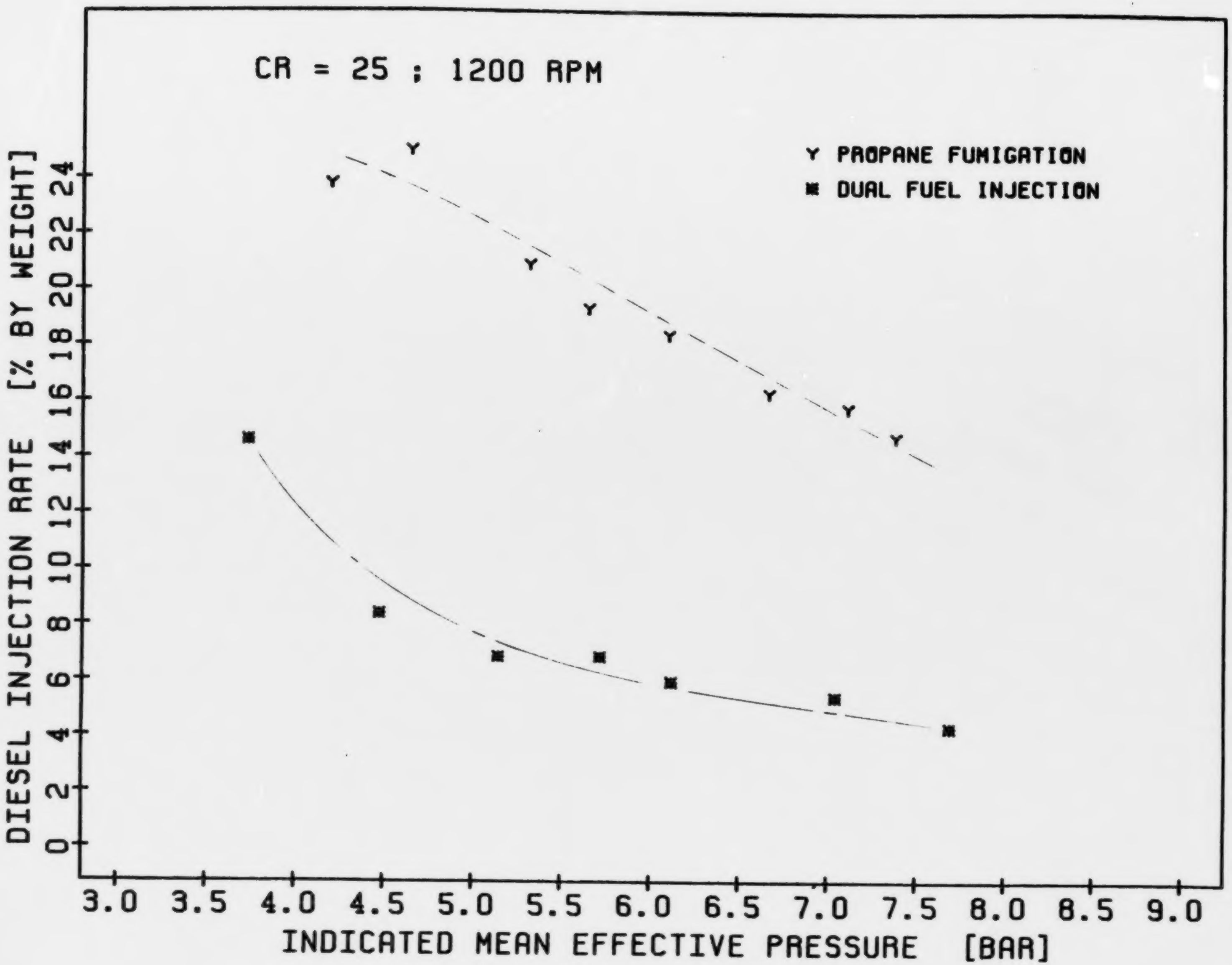

CR = 25 ; 1200 RPM
Y PROPANE FUMIGATION
✱ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-16

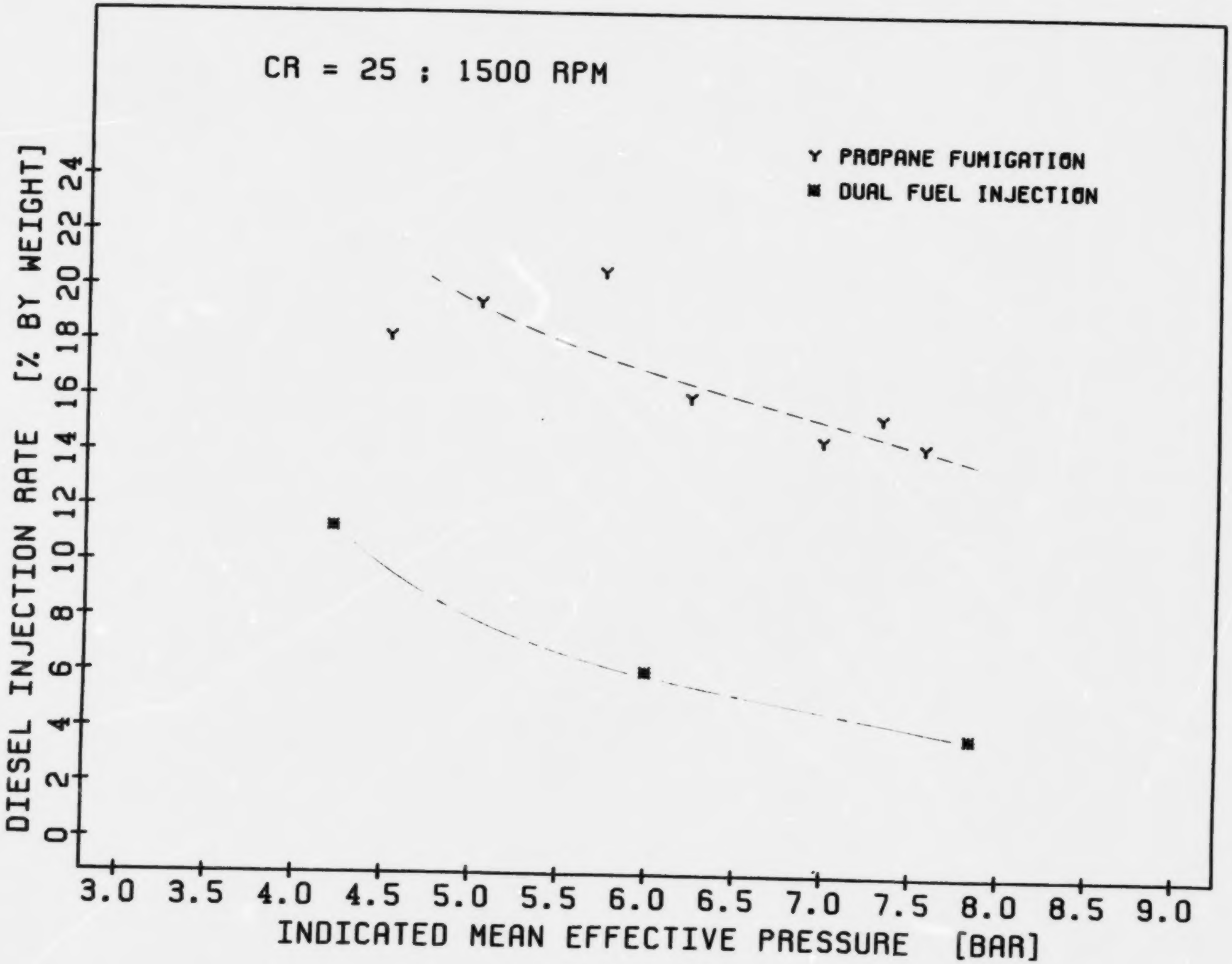

CR = 25 ; 1500 RPM
Y PROPANE FUMIGATION
■ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-17

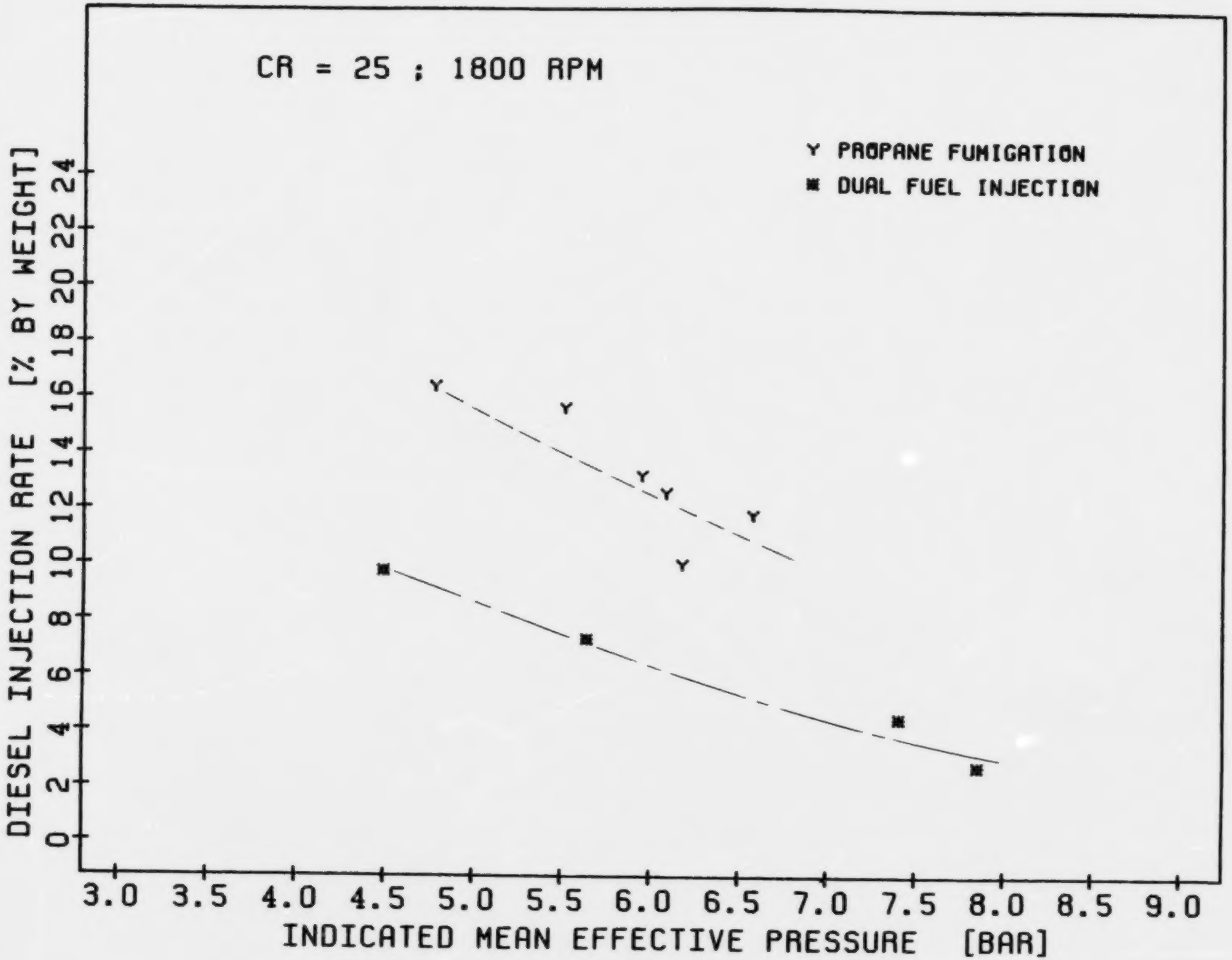

CR = 25 ; 1800 RPM
Y PROPANE FUMIGATION
✱ DUAL FUEL INJECTION
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-18

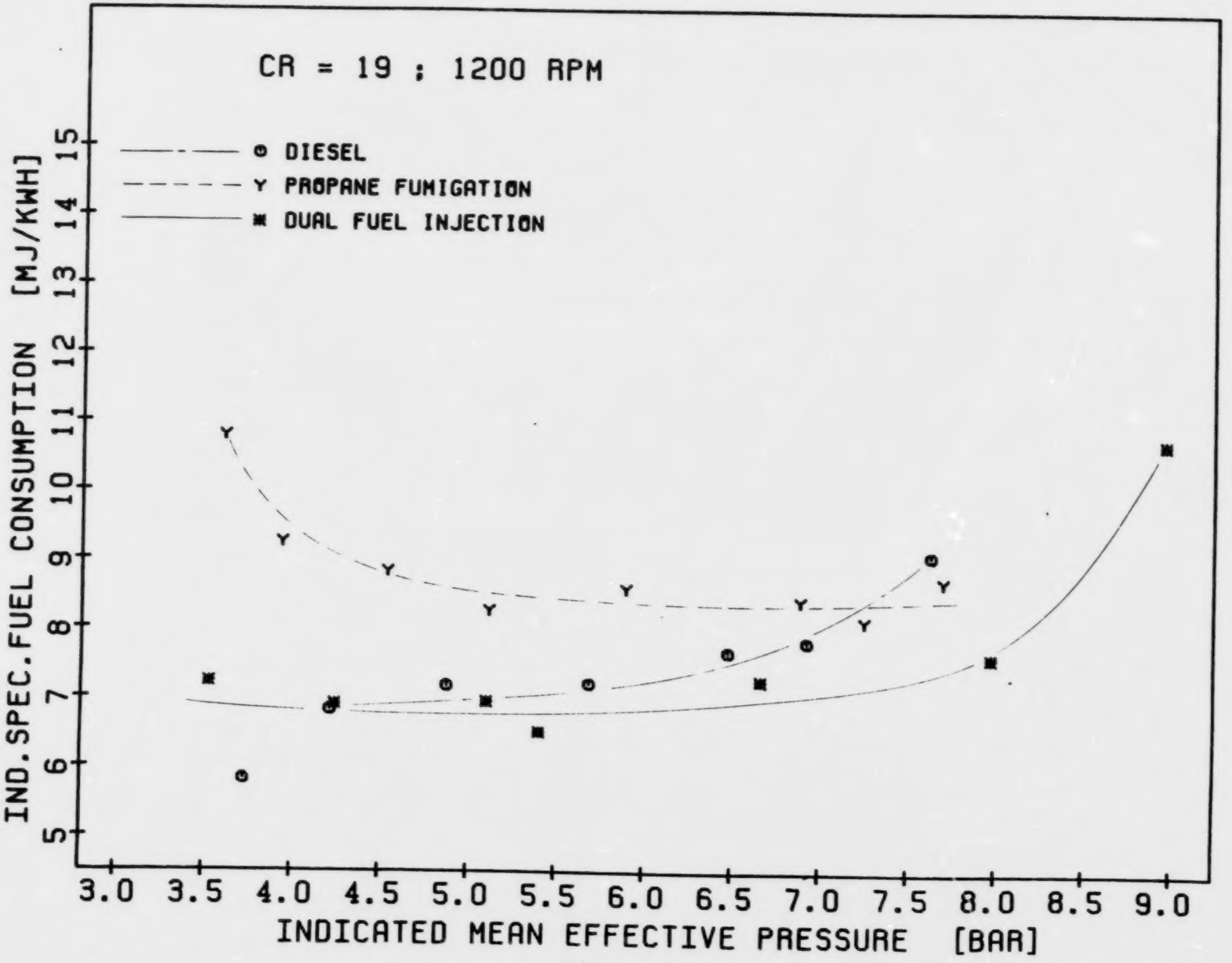

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-19

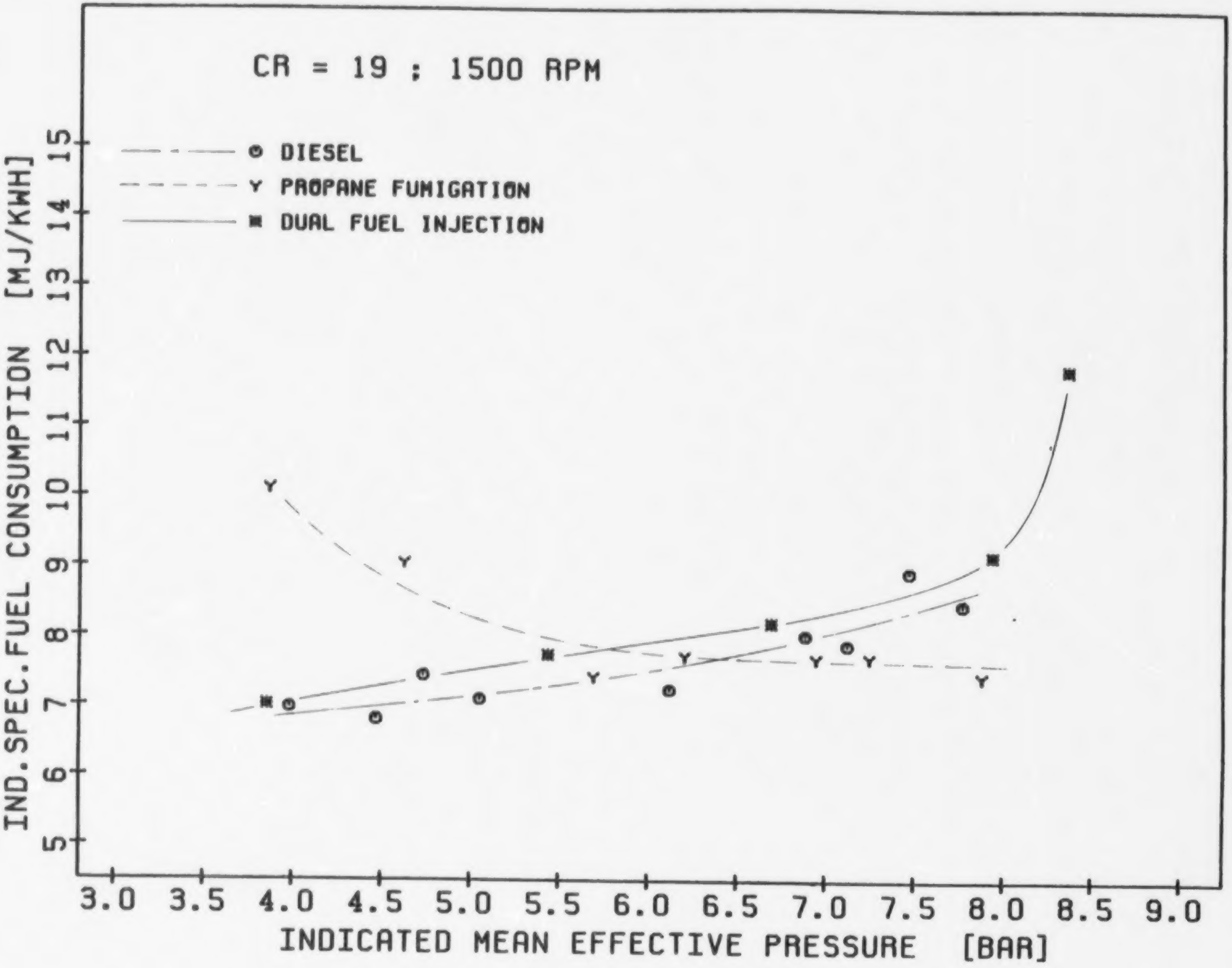

CR = 19 : 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-20

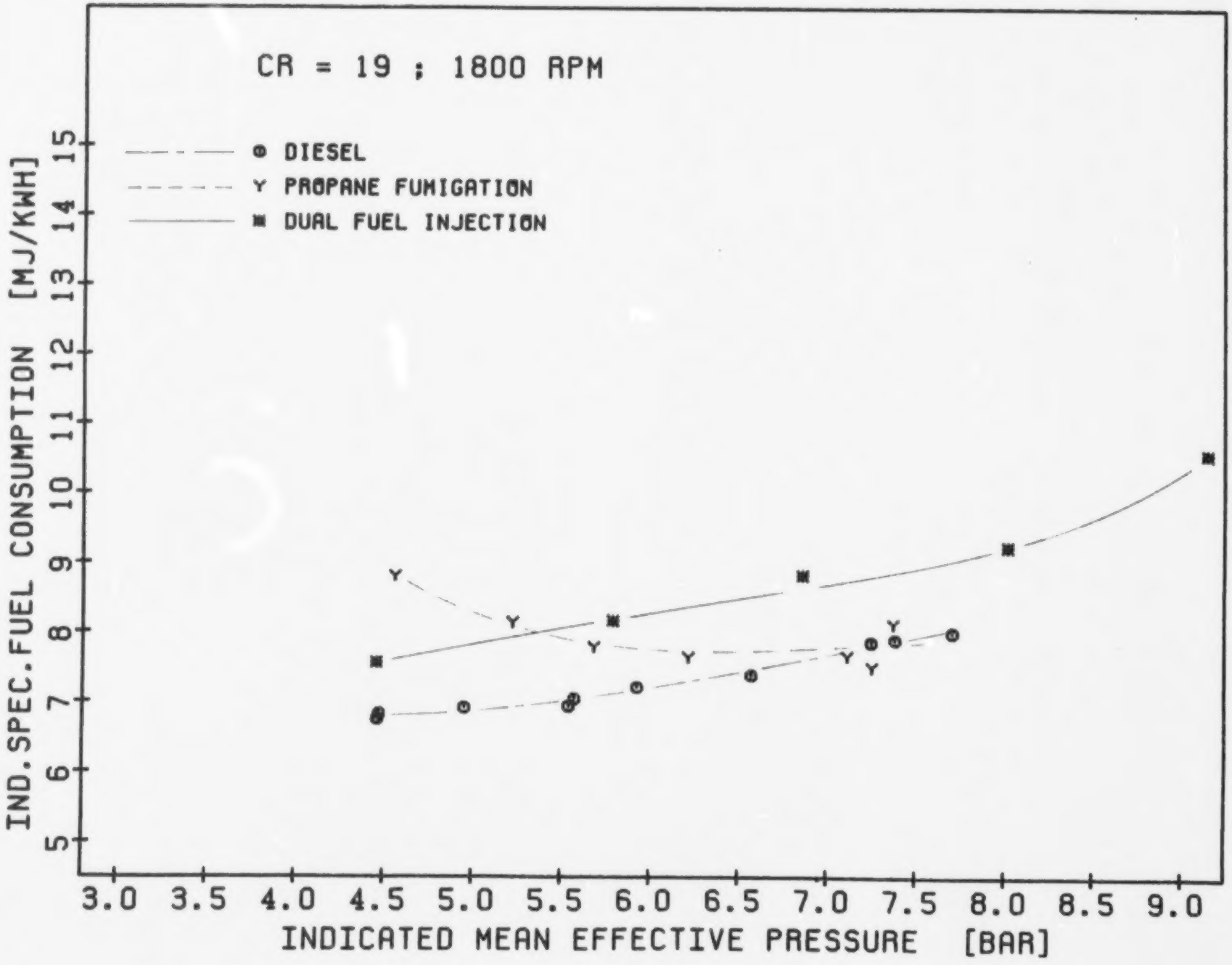

CR = 19 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-21

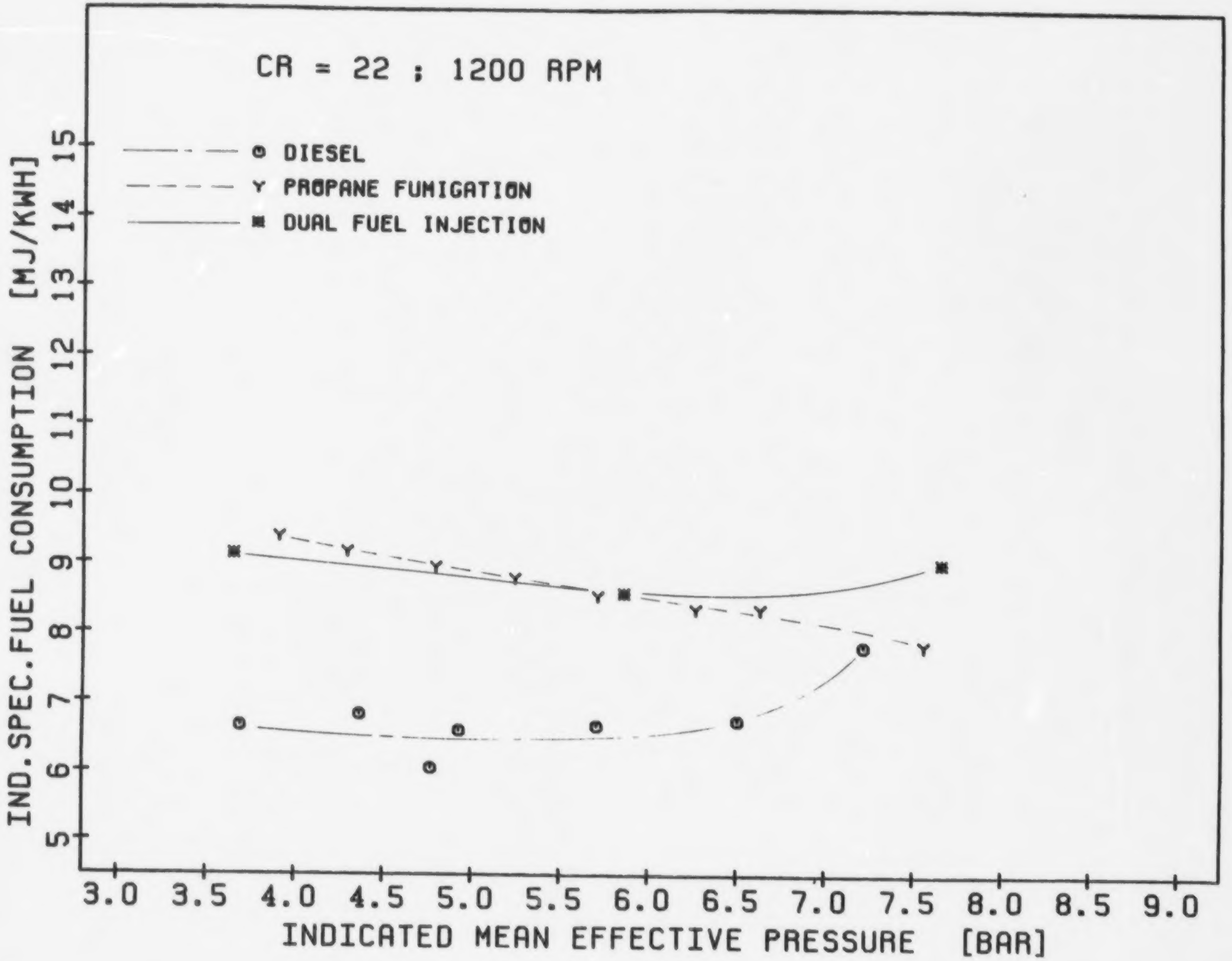

CR = 22 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-22

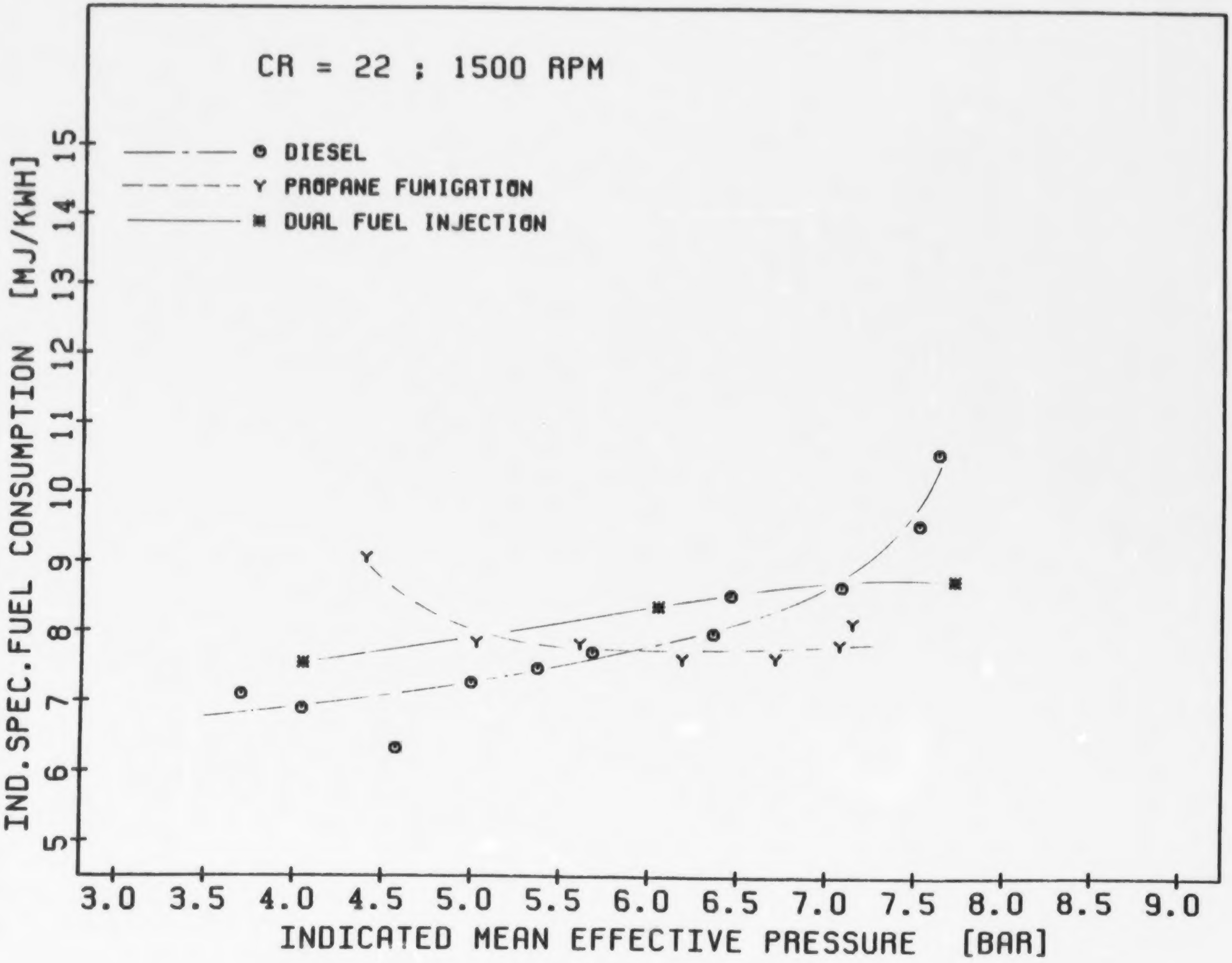

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-23

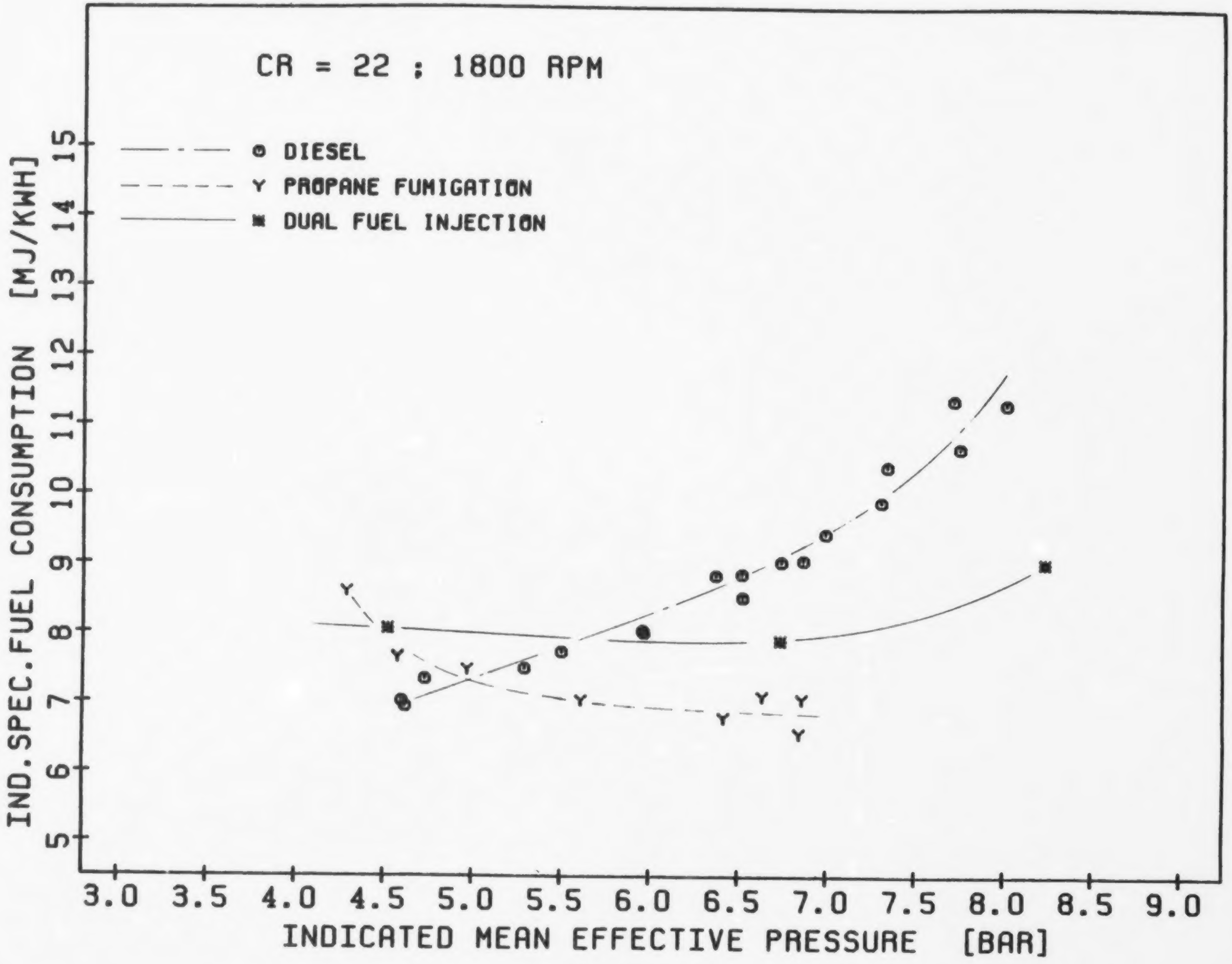

CR = 22 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-24

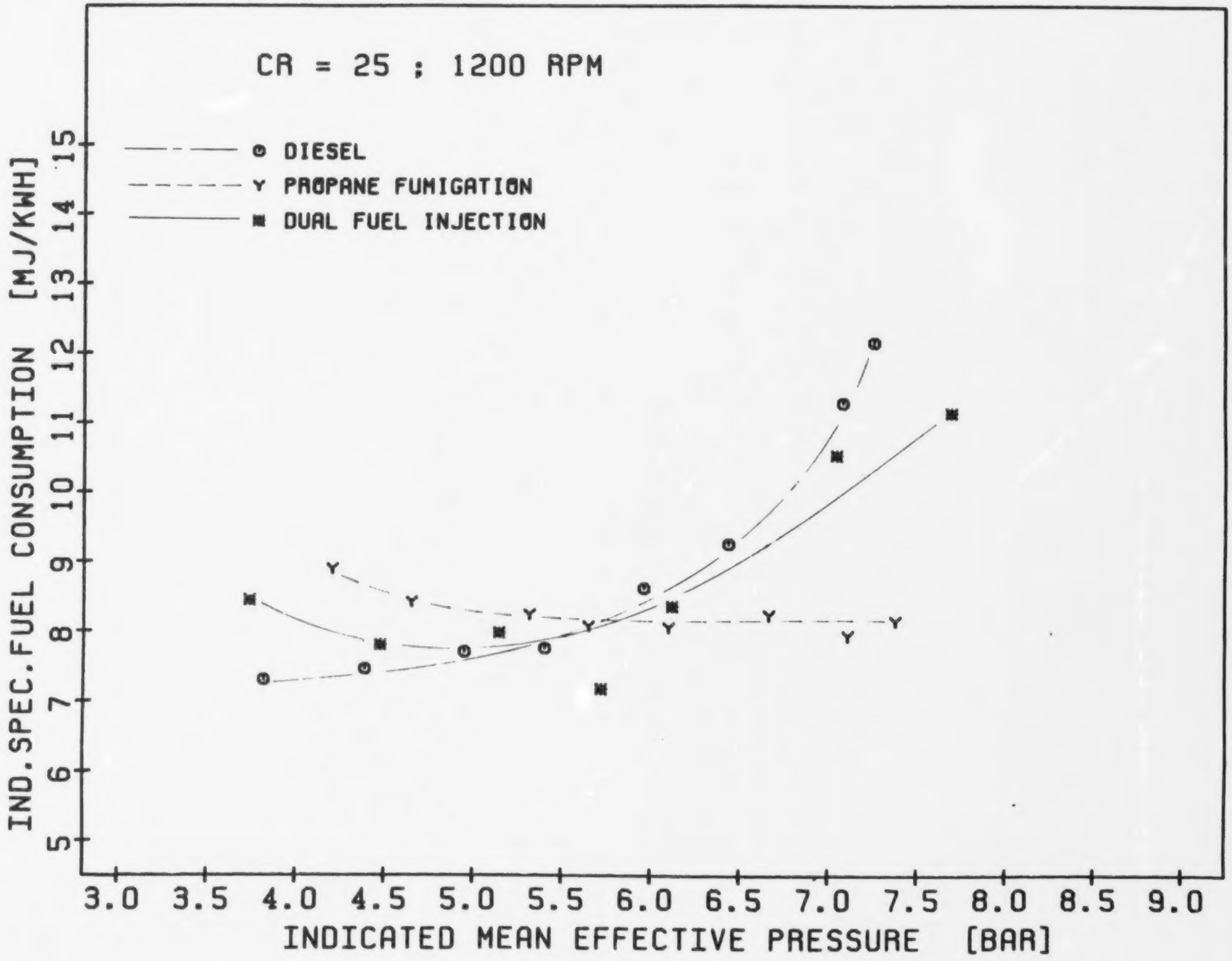

CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-25

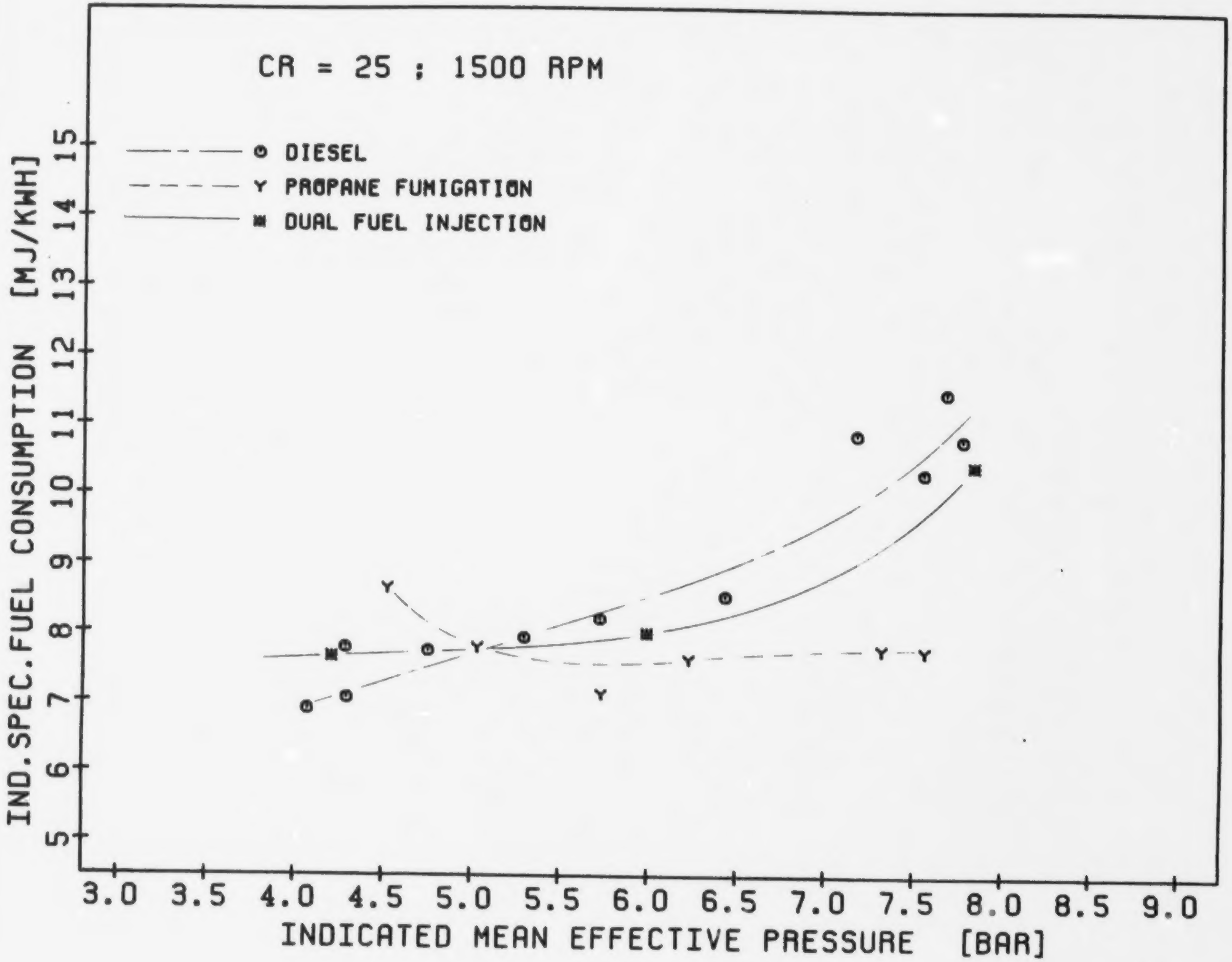

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-26

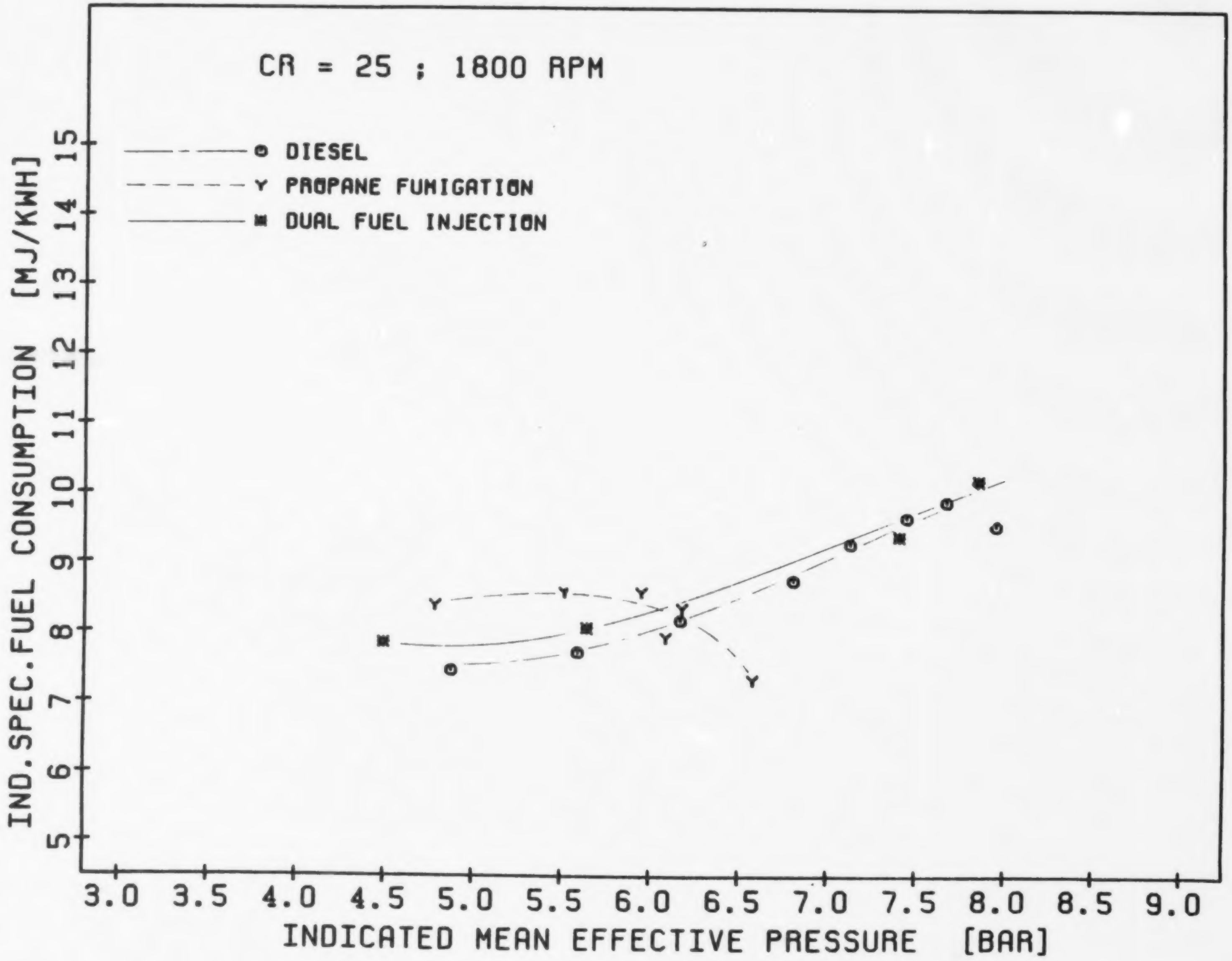

CR = 25 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-27

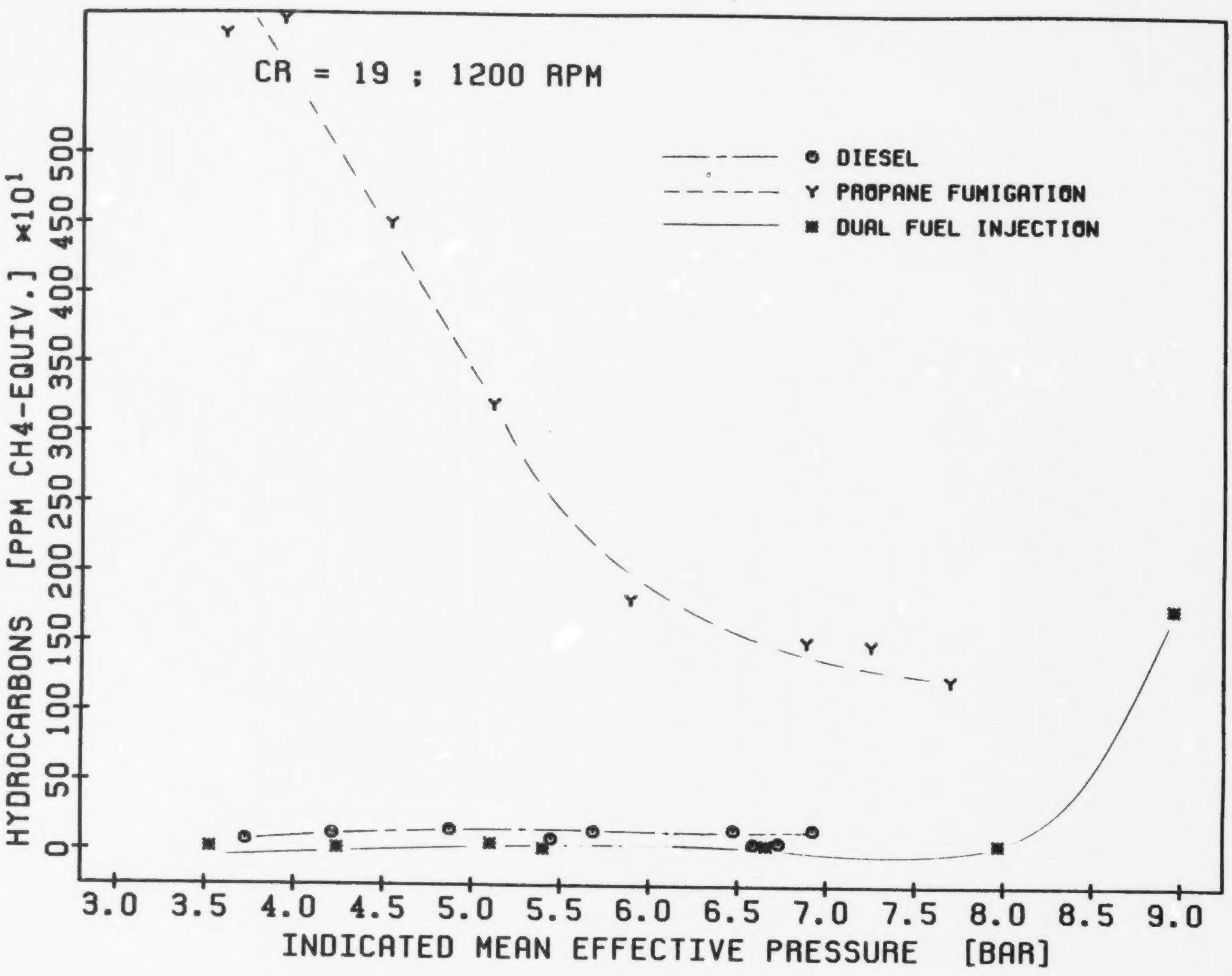

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-28

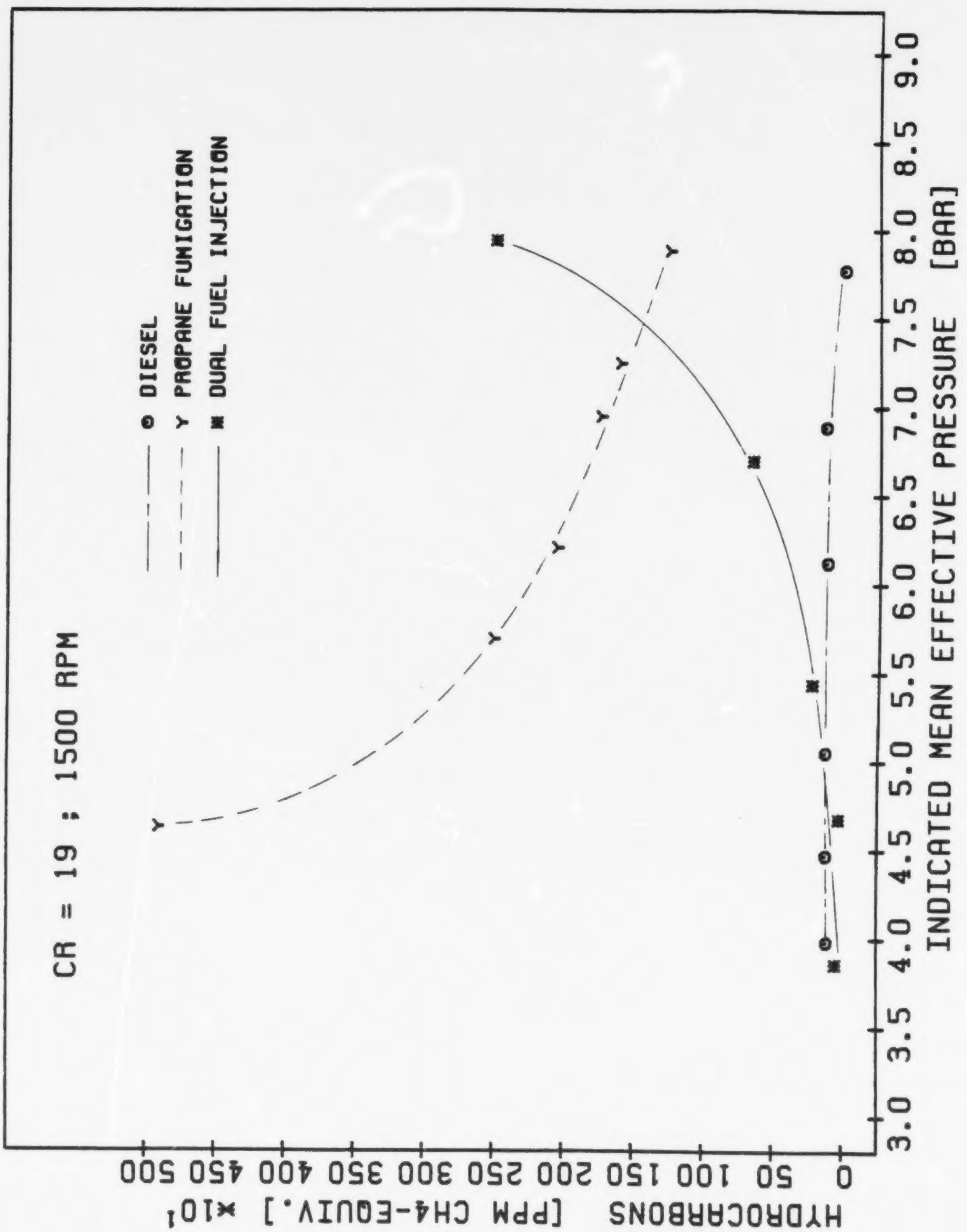

CR = 19 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0

Figure 4-29

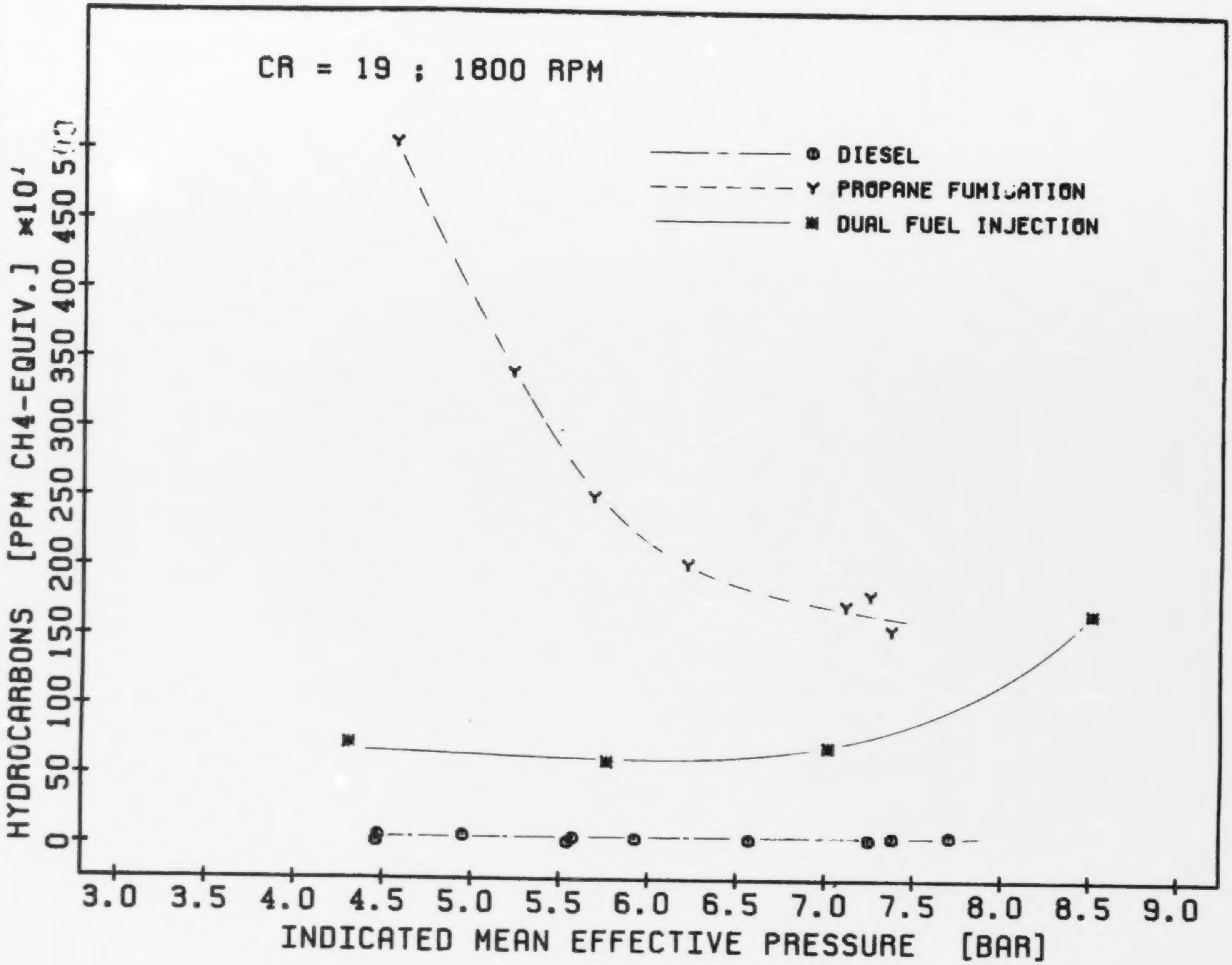

CR = 19 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-30

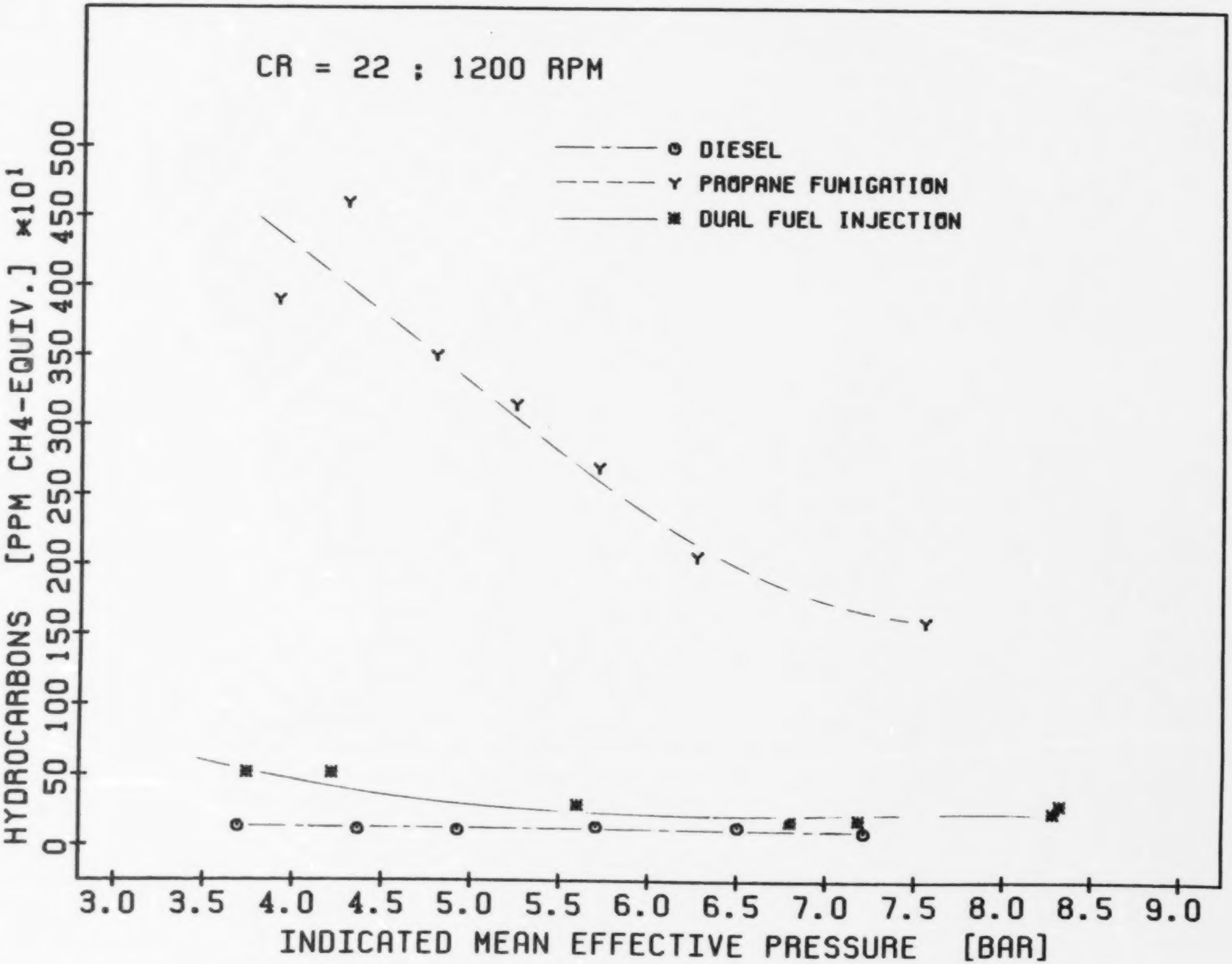

CR = 22 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] $\times 10^1$
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-31

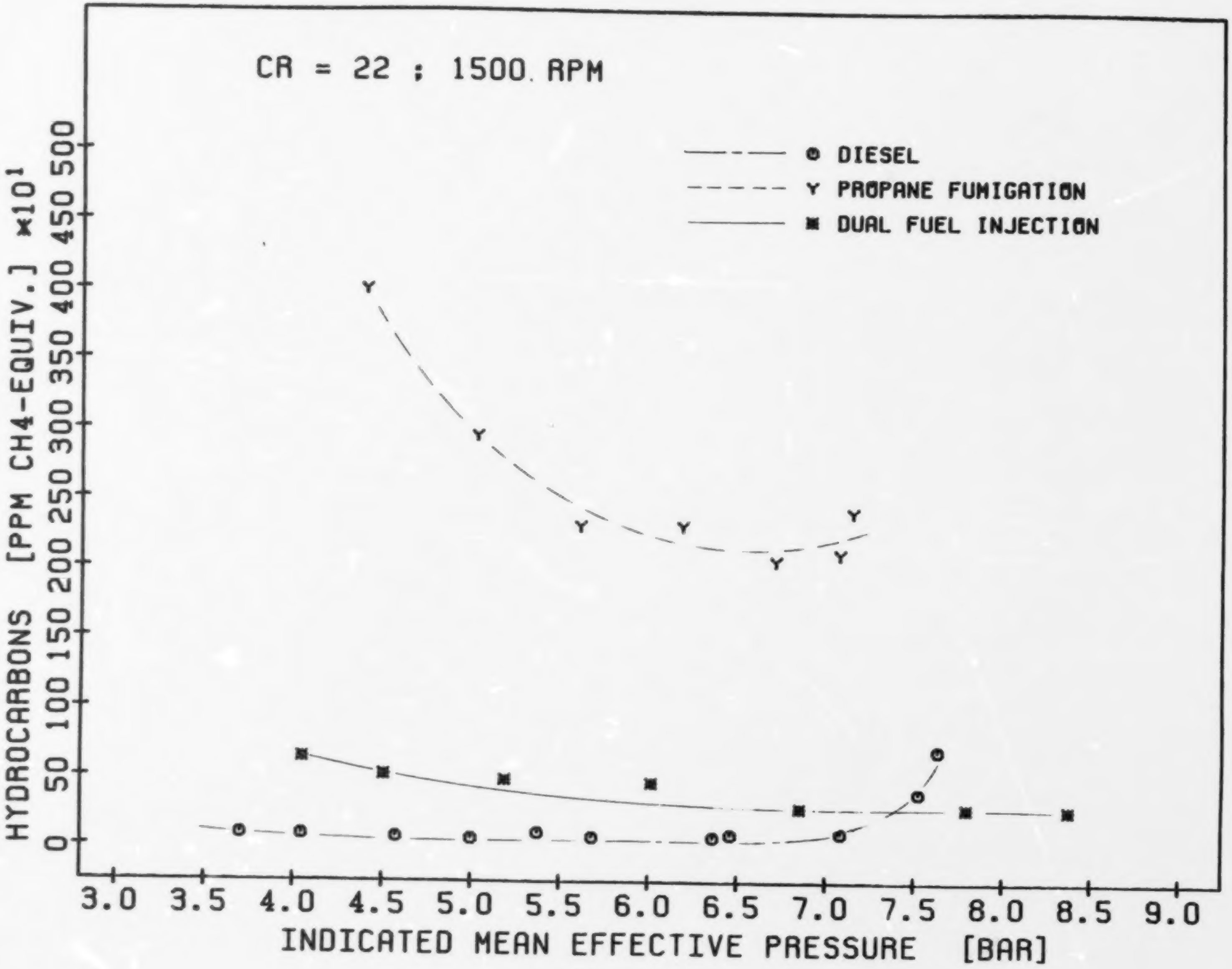

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-32

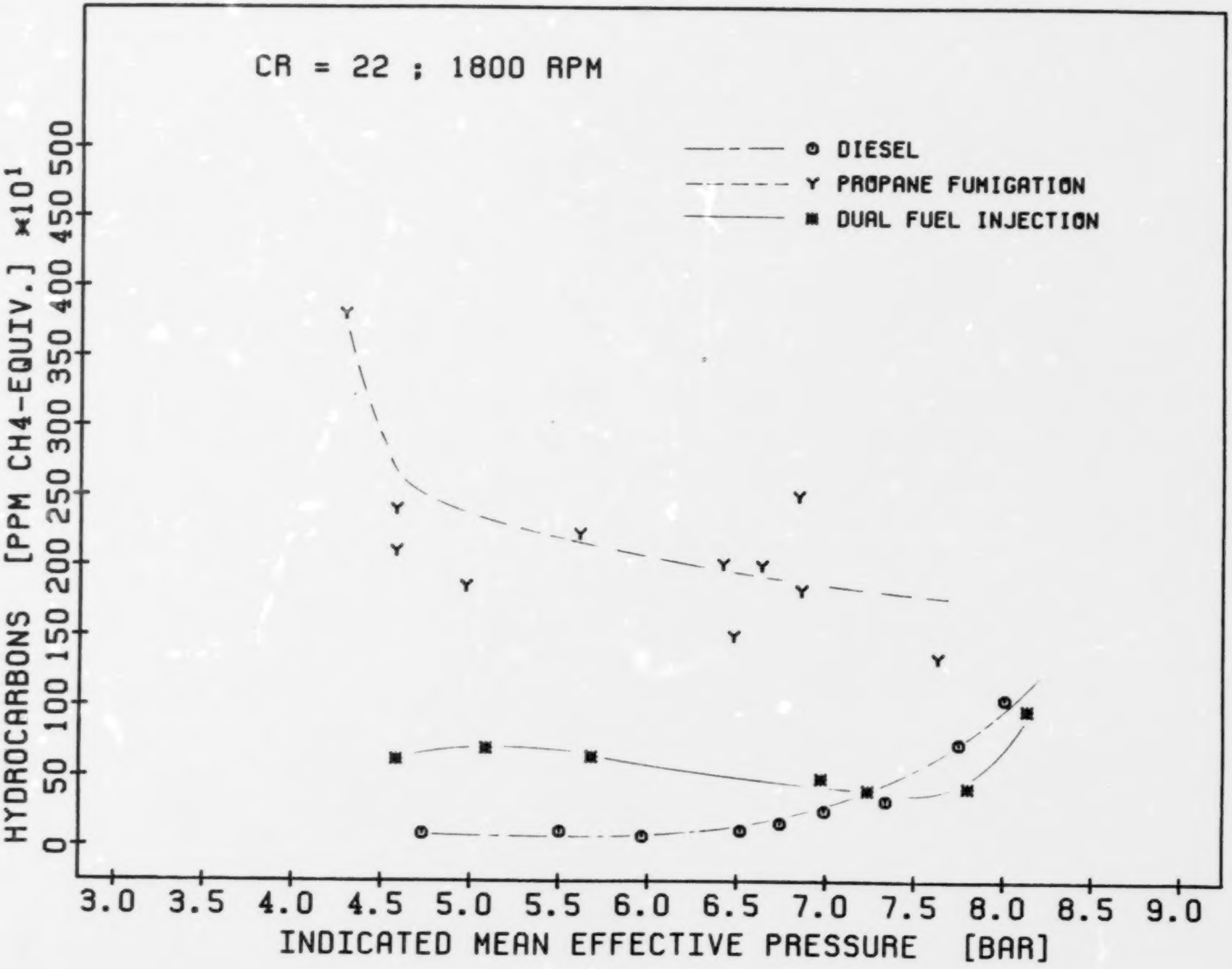

CR = 22 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-33

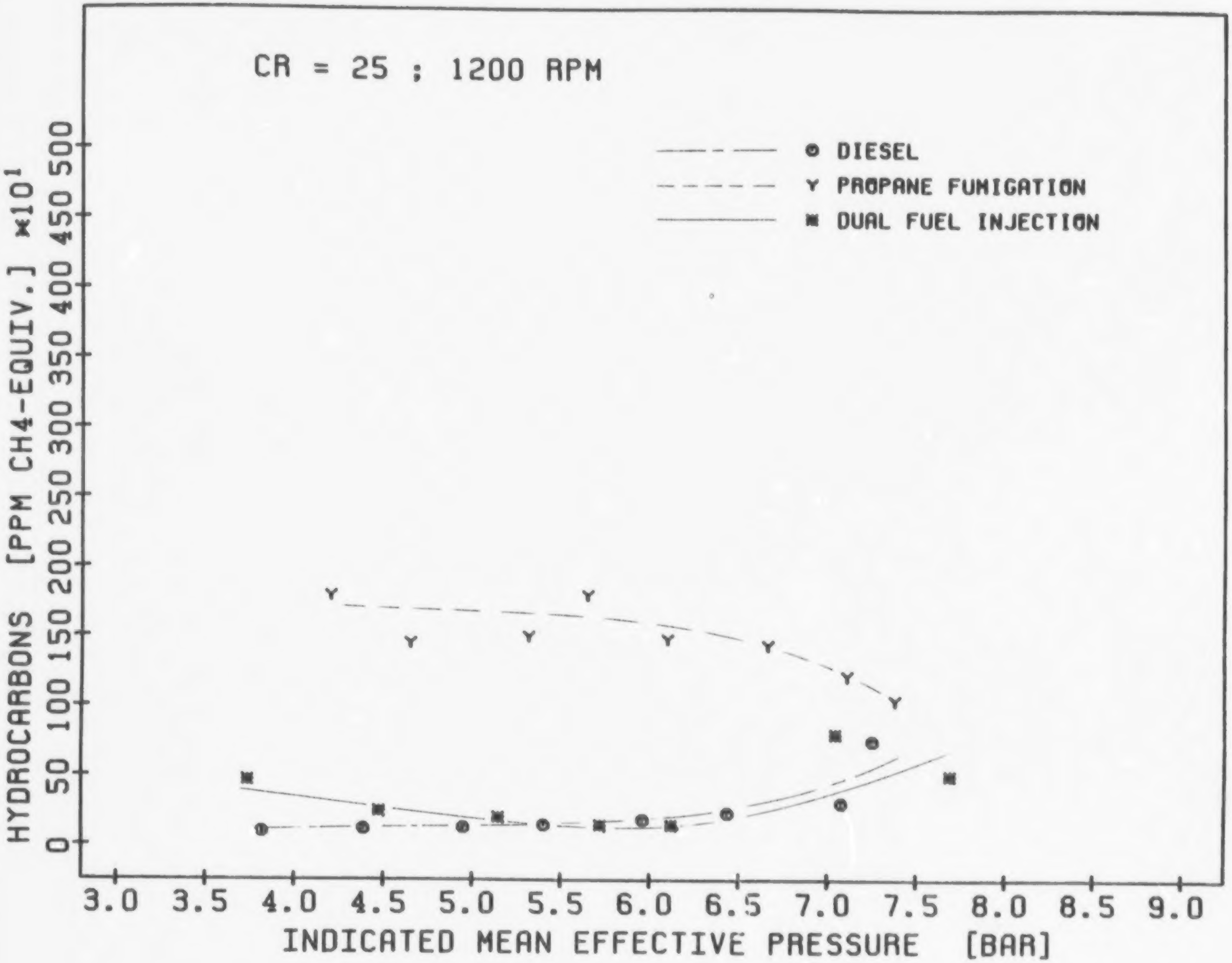

CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-34

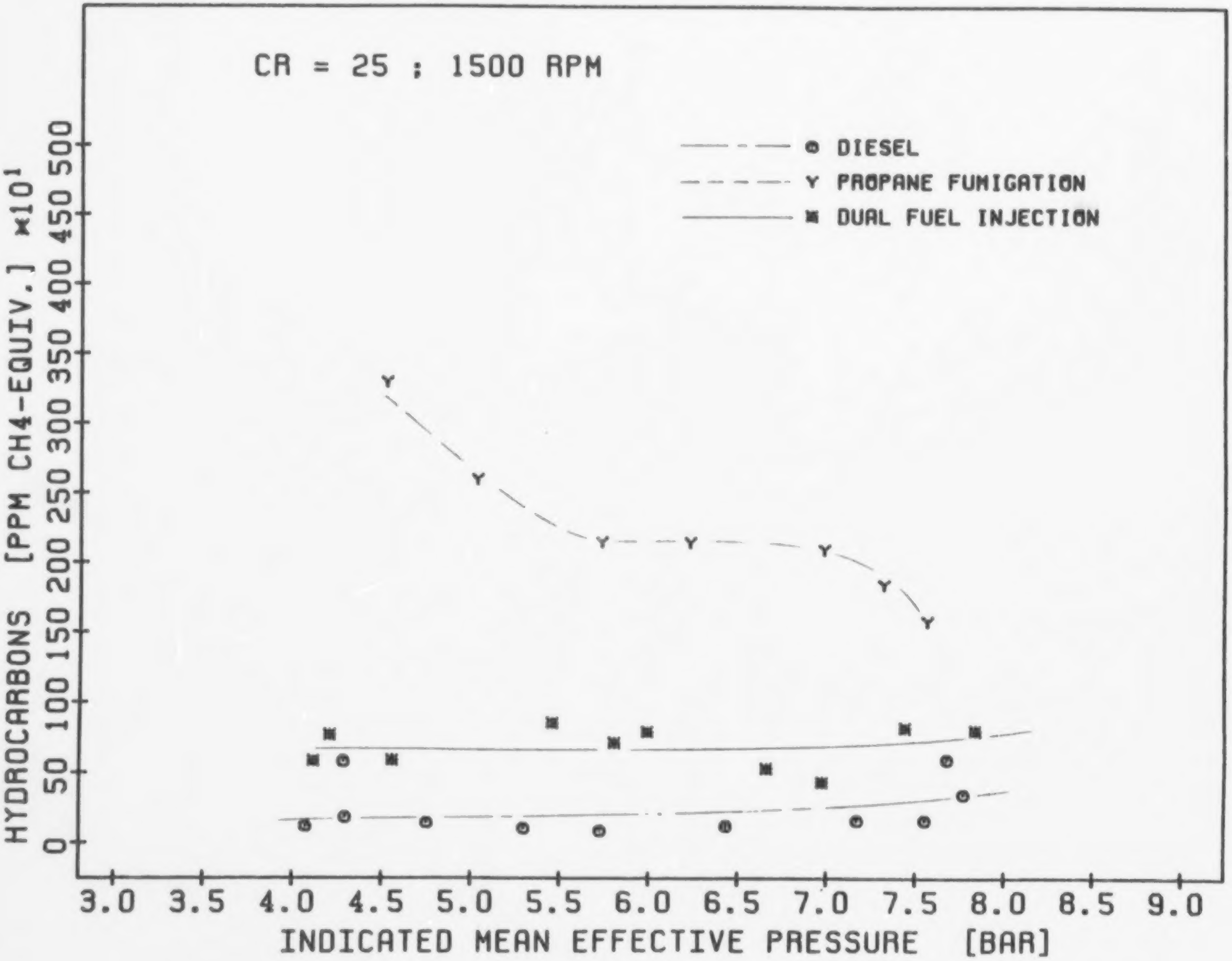

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-35

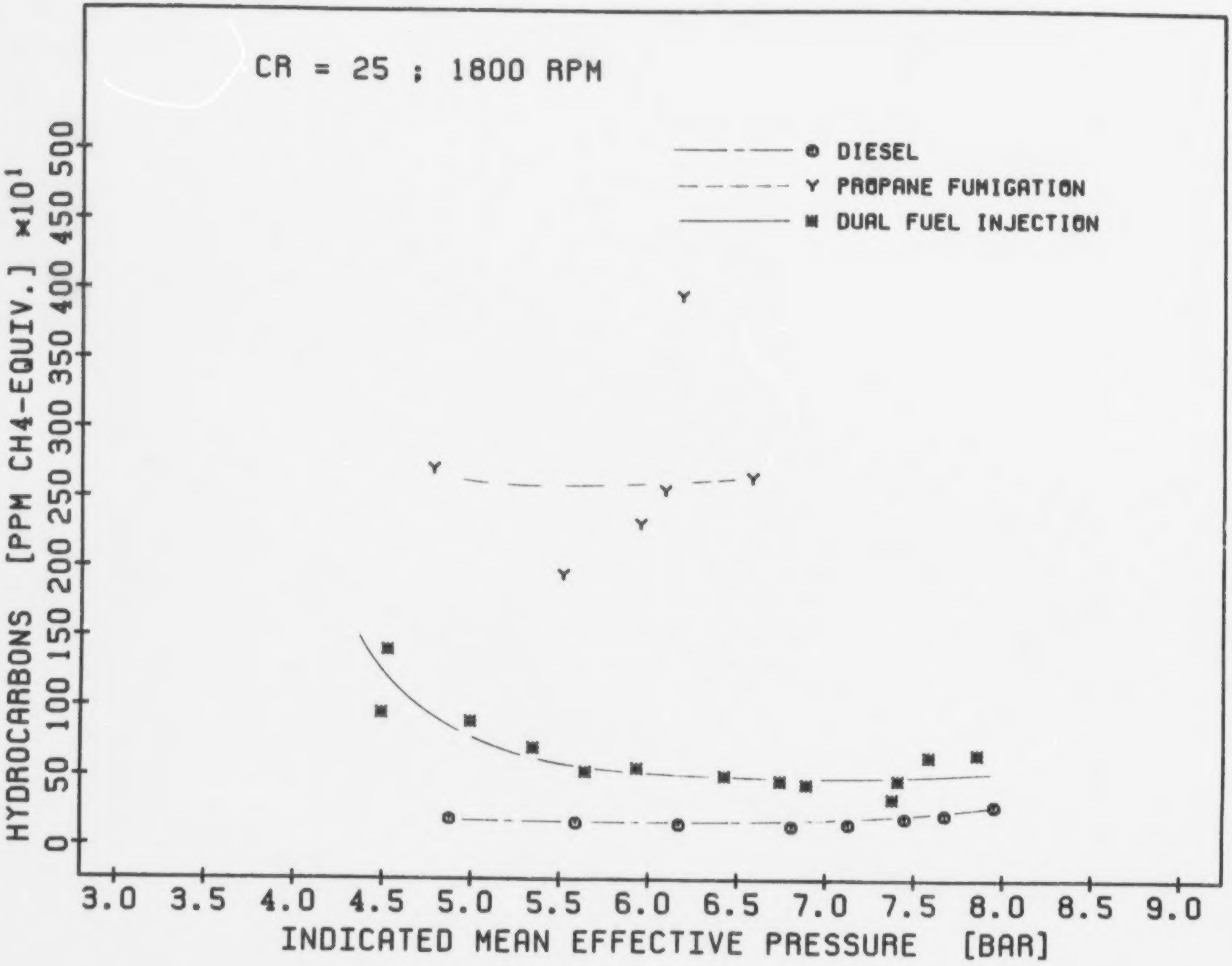

CR = 25 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
450
500
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-36

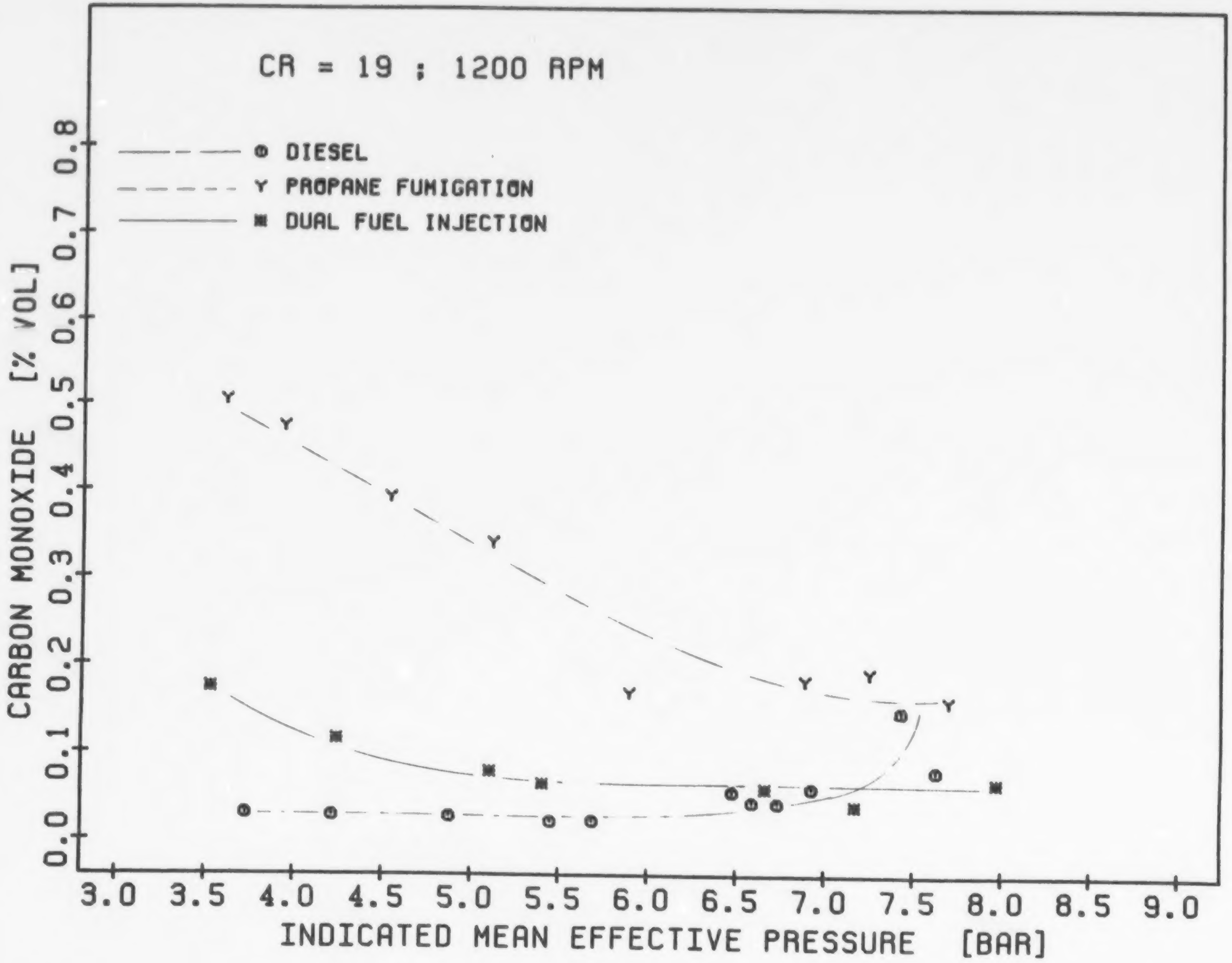

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-37

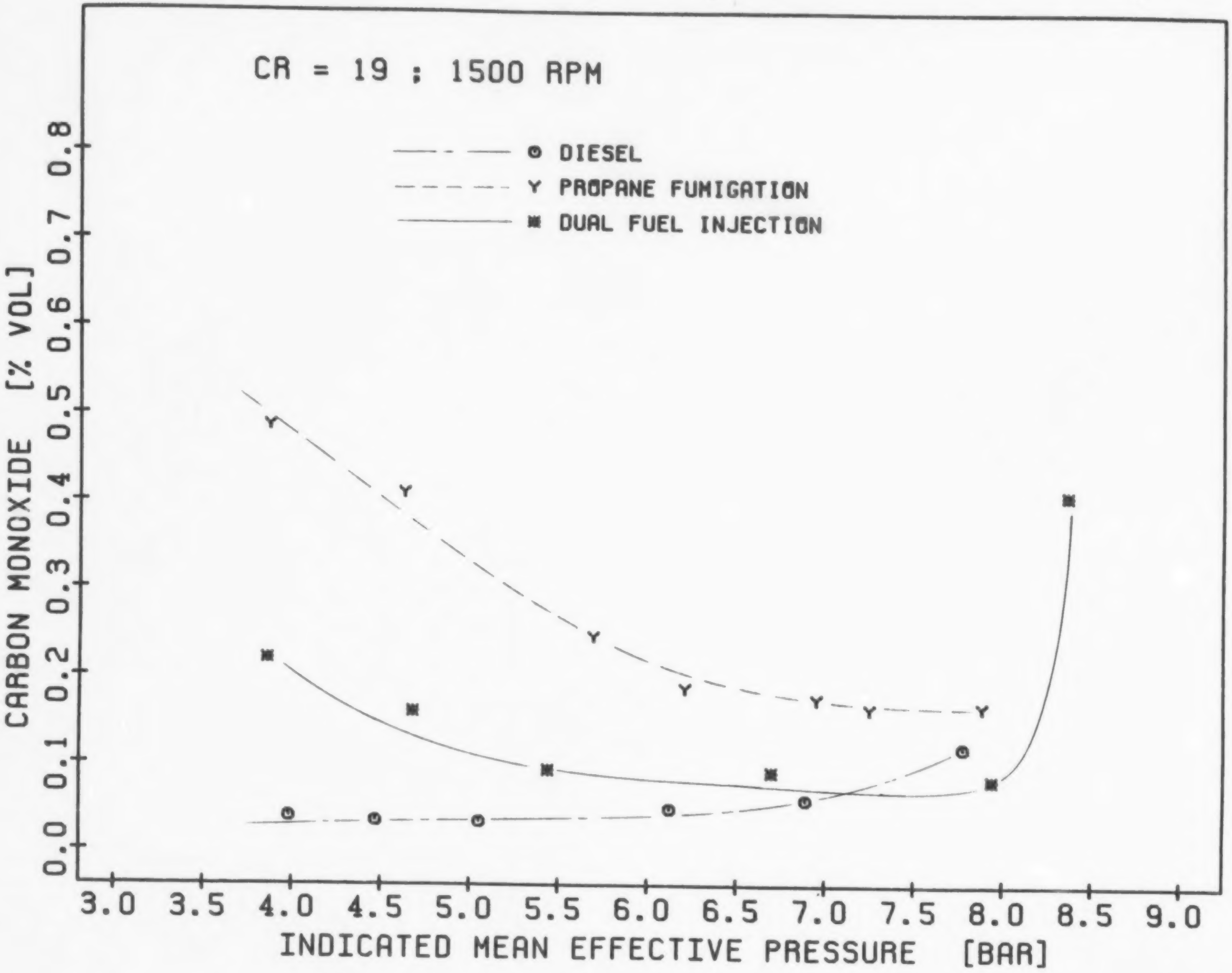

CR = 19 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-38

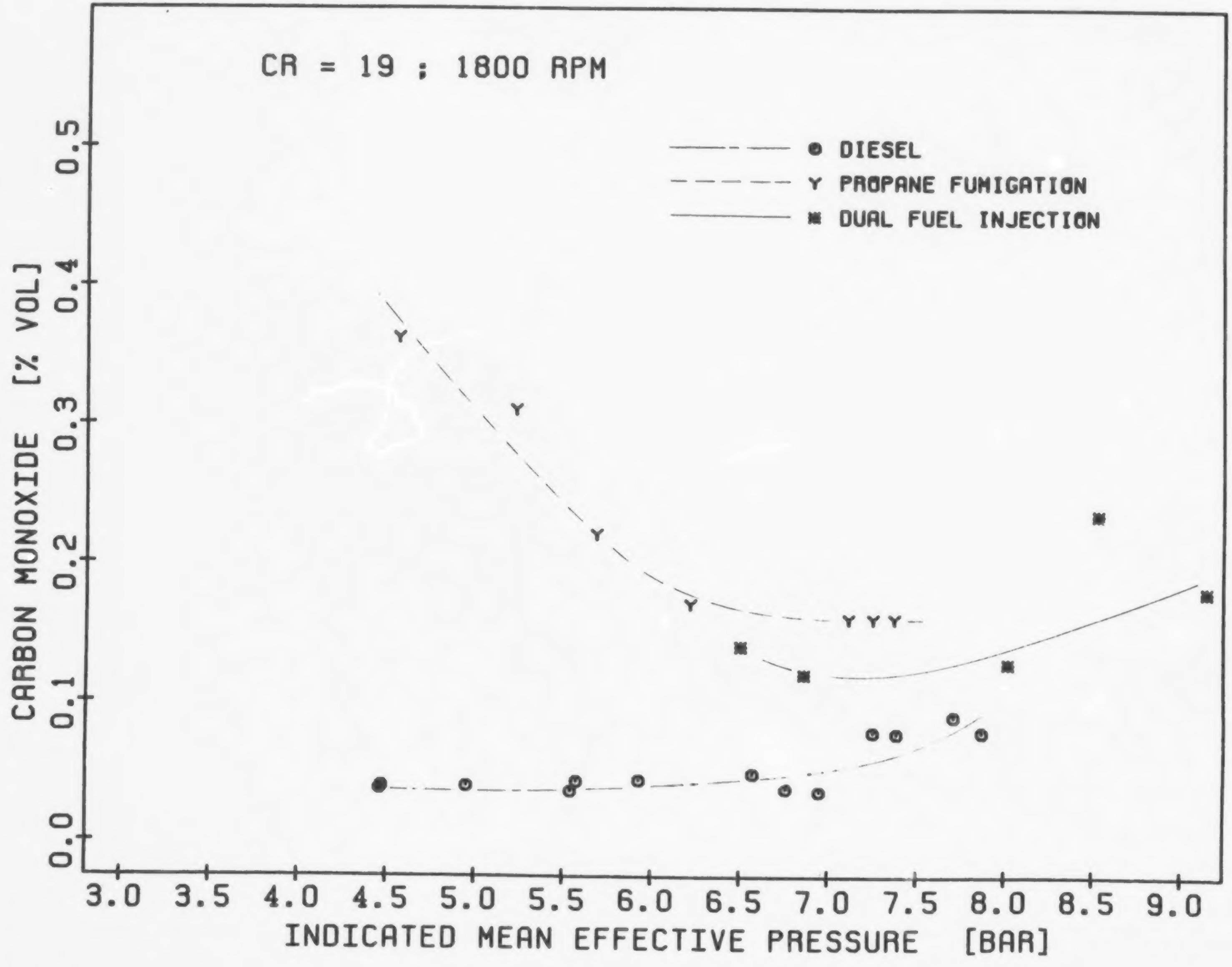

CR = 19 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.1
0.2
0.3
0.4
0.5
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-39

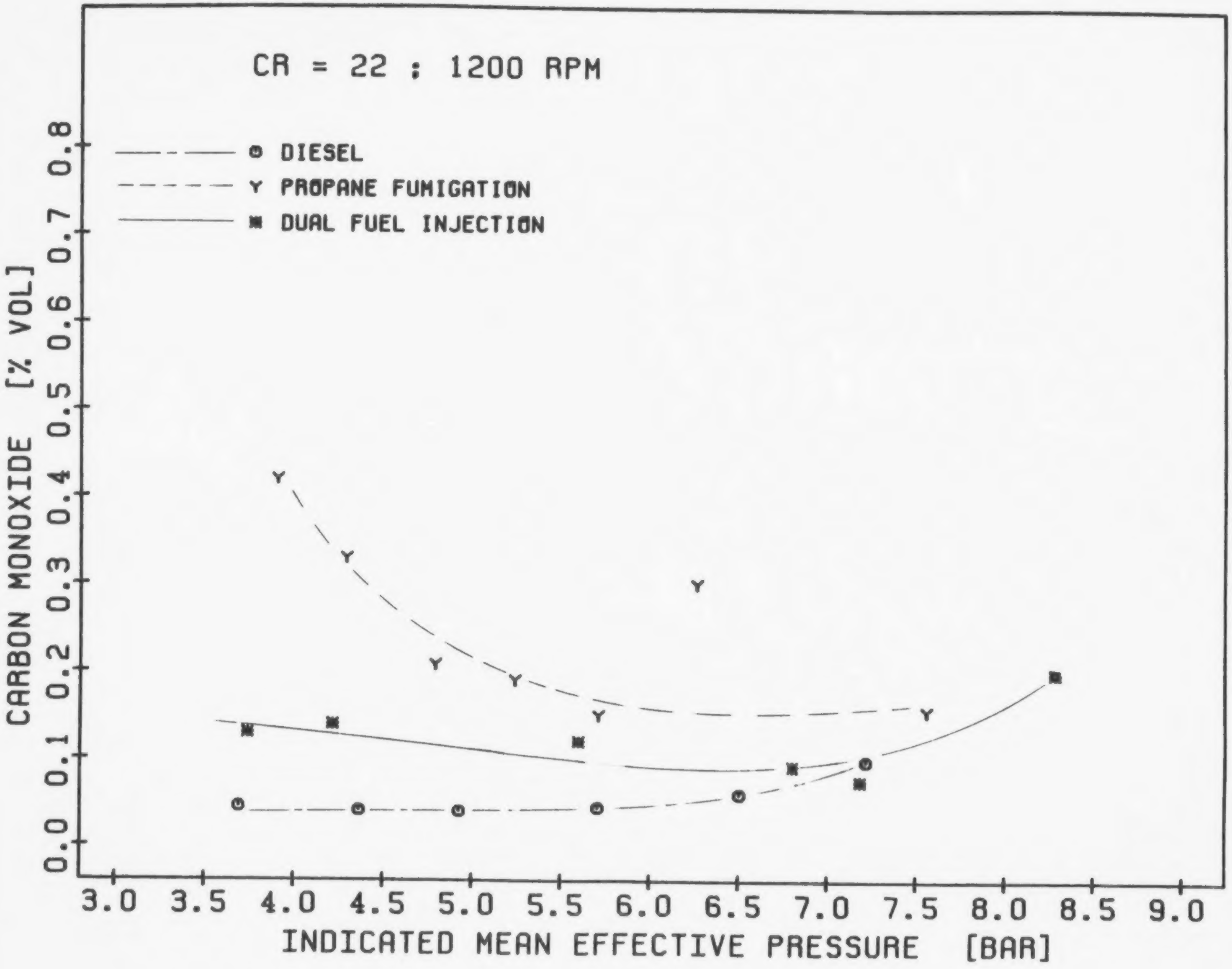

CR = 22 : 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-40

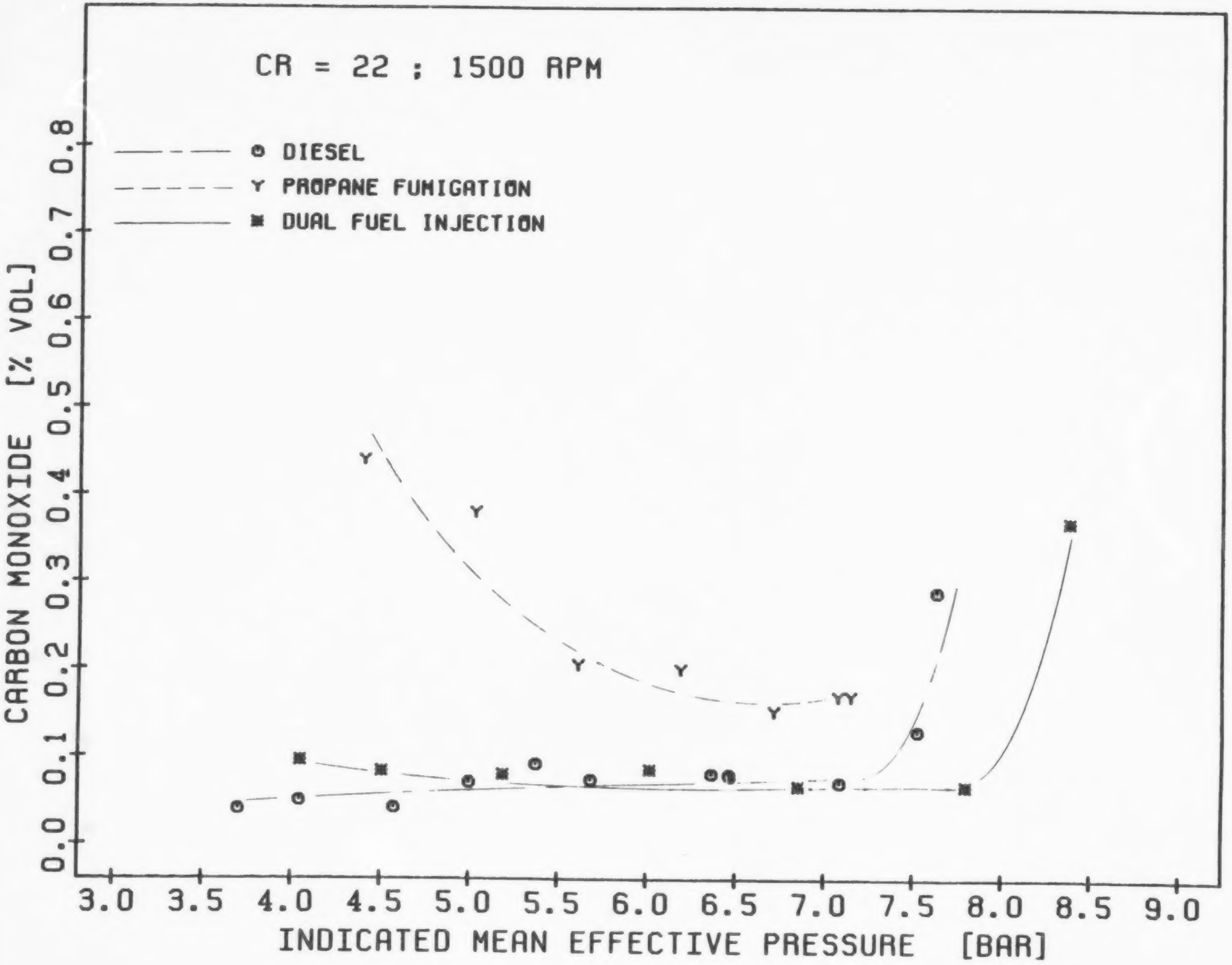

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-41

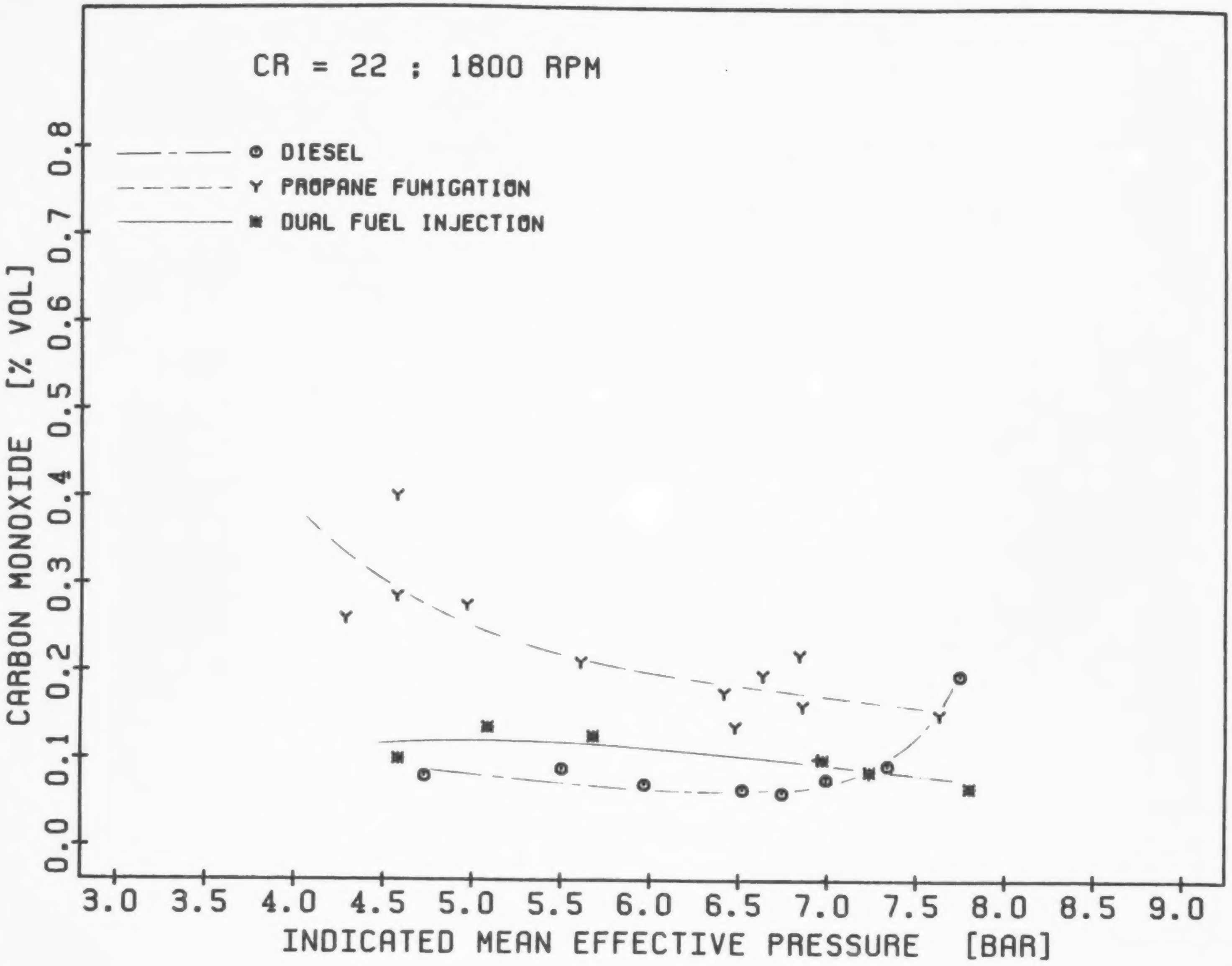

CR = 22 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-42

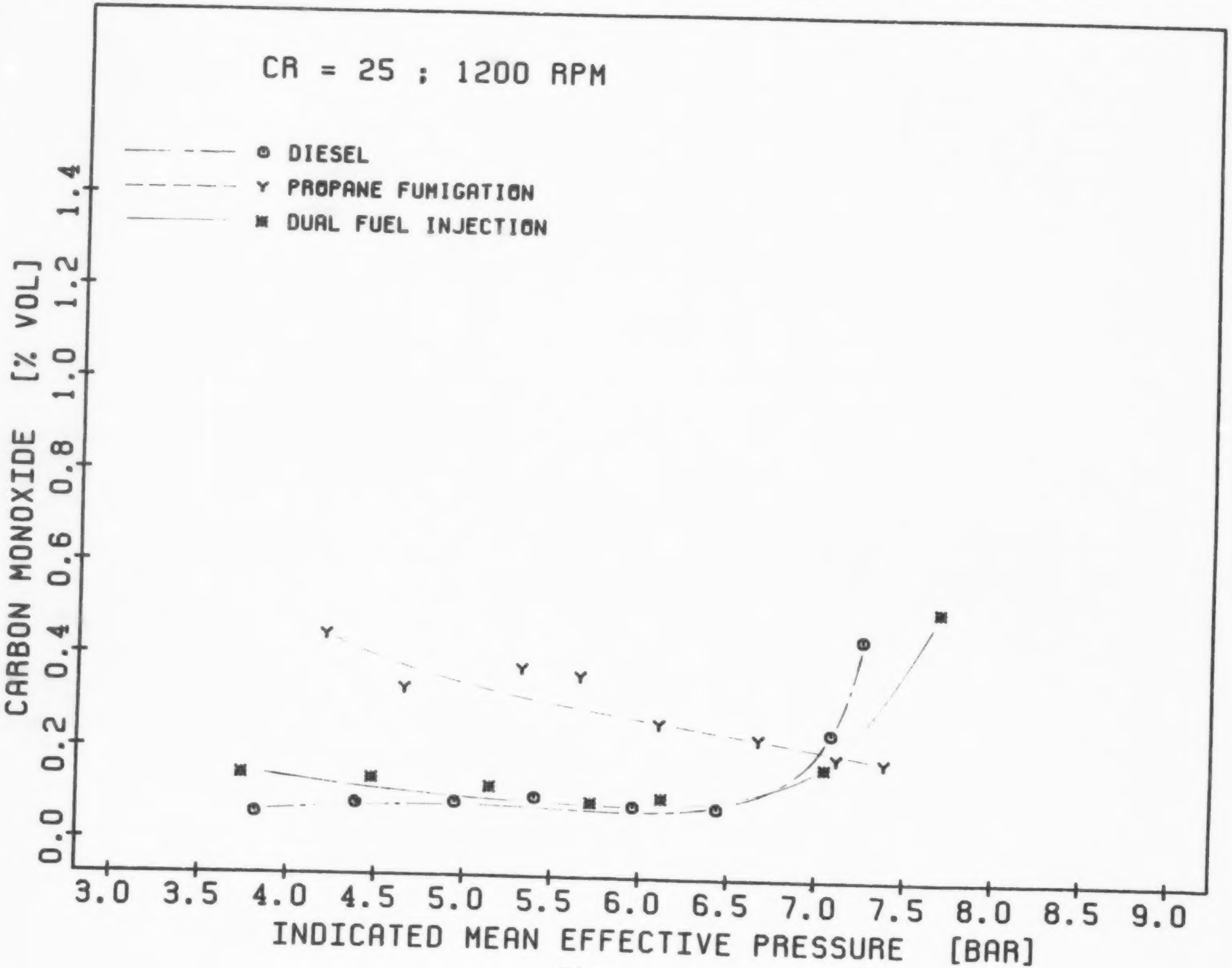

CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-43

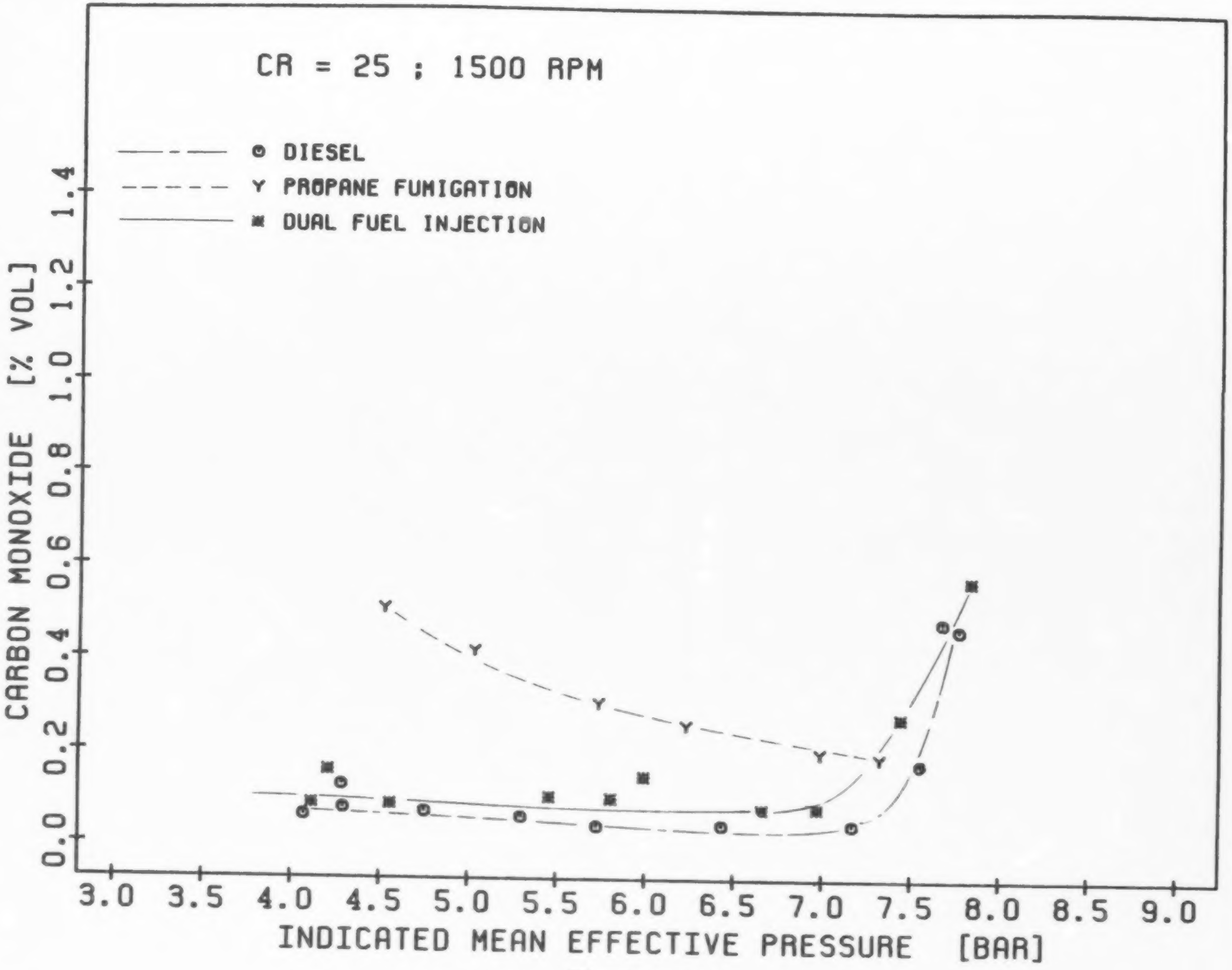

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-44

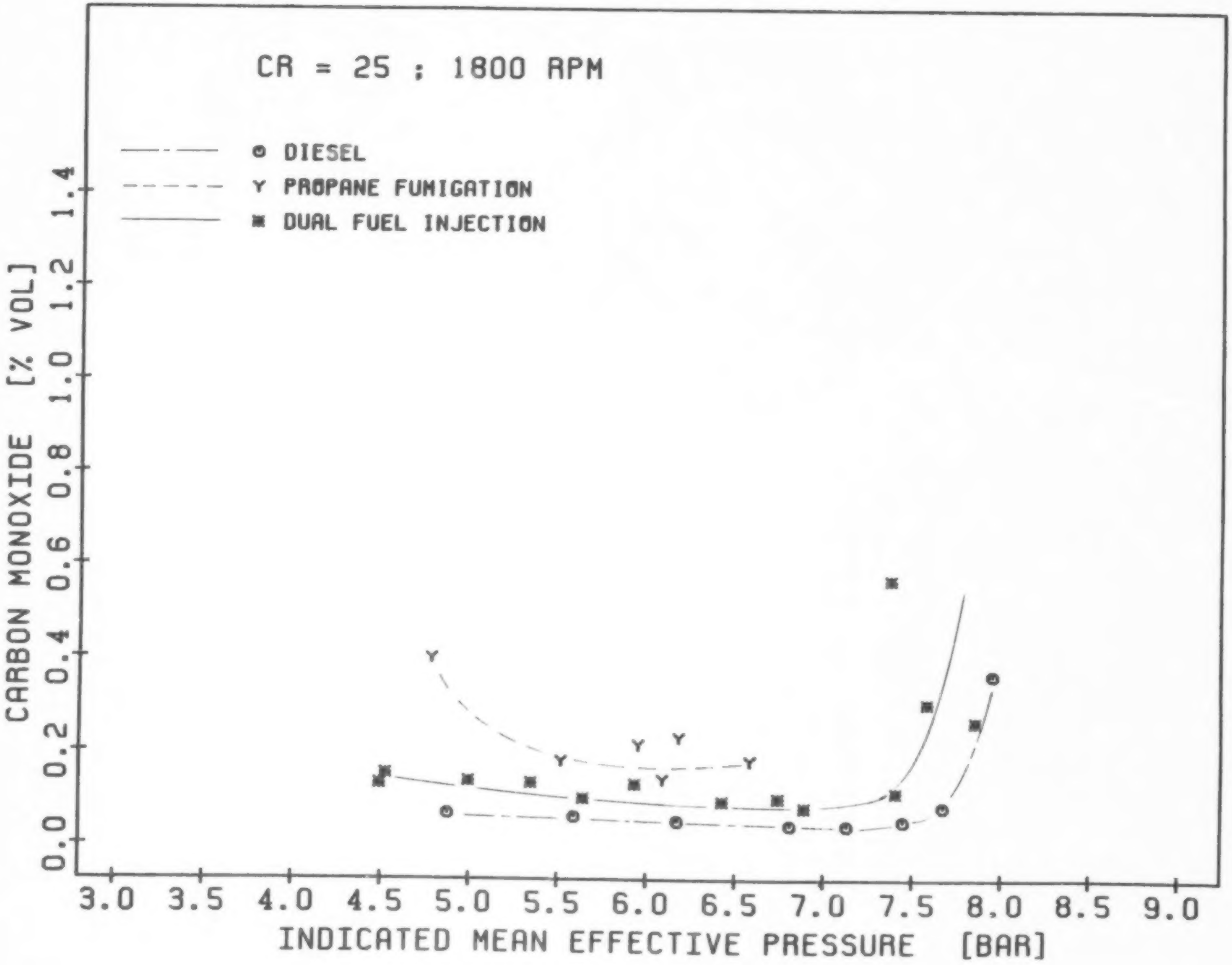

CR = 25 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
CARBON MONOXIDE [% VOL]
0.0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-45

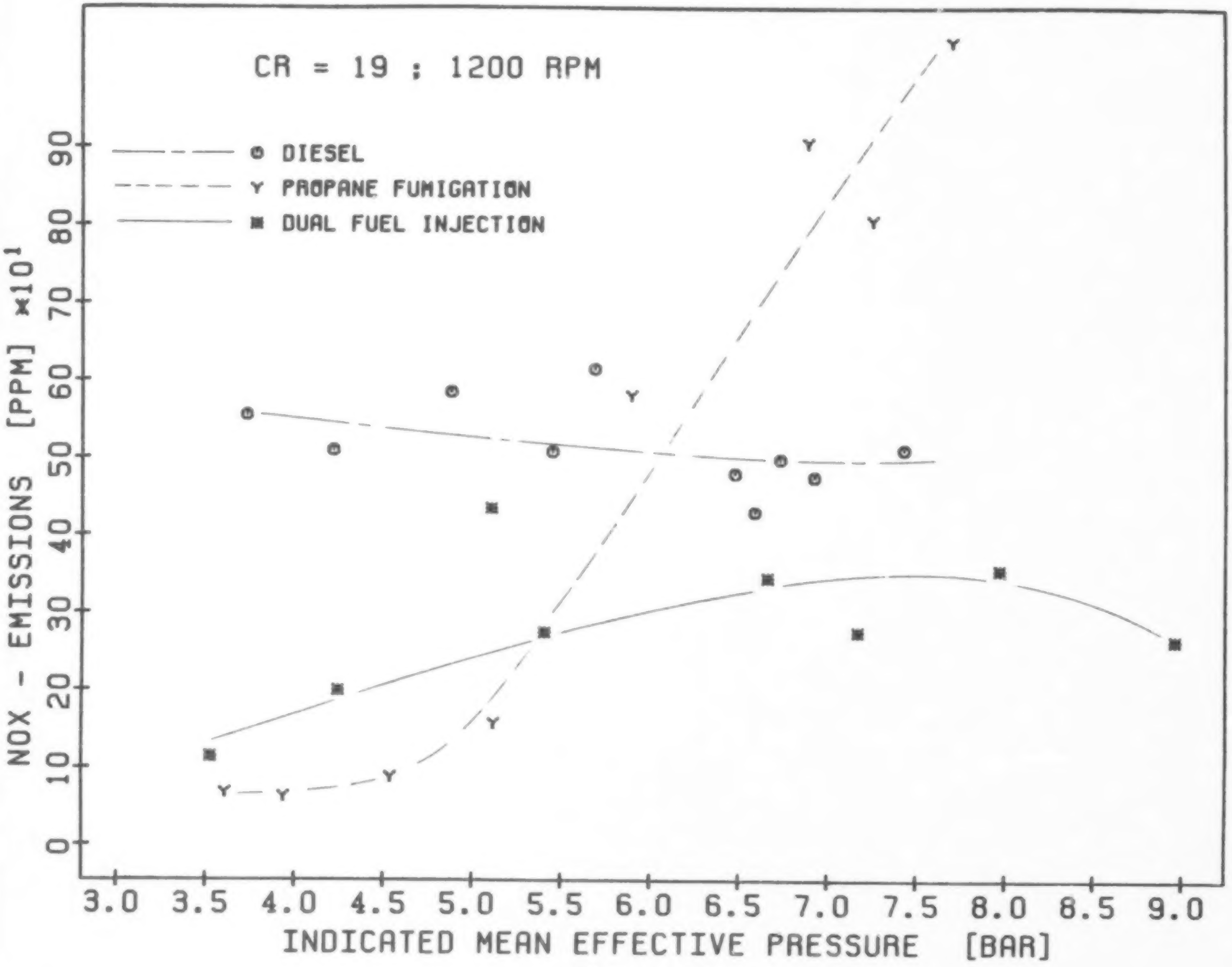

CR = 19 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] ×10¹
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-46

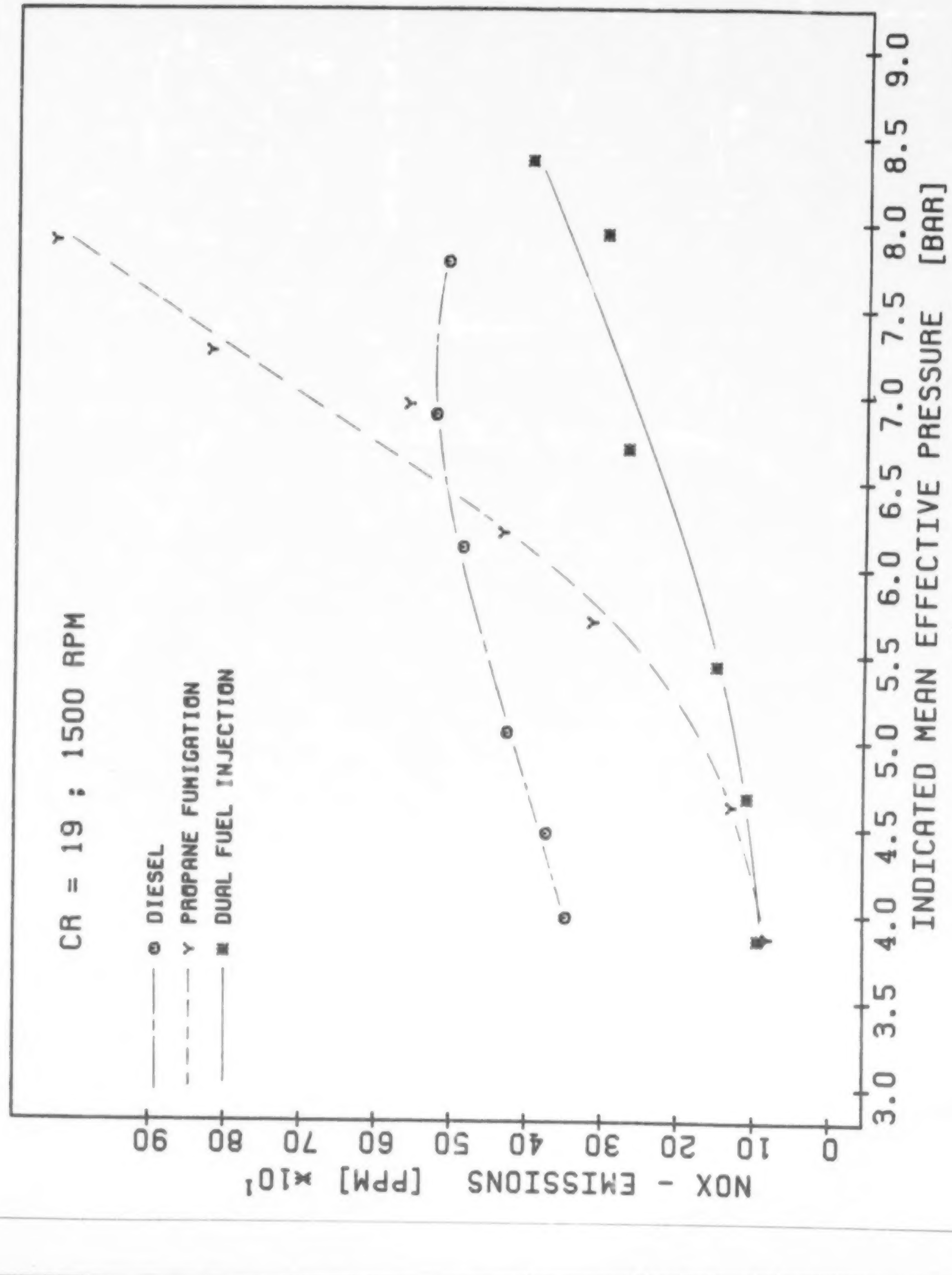

CR = 19 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] ×10¹
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-47

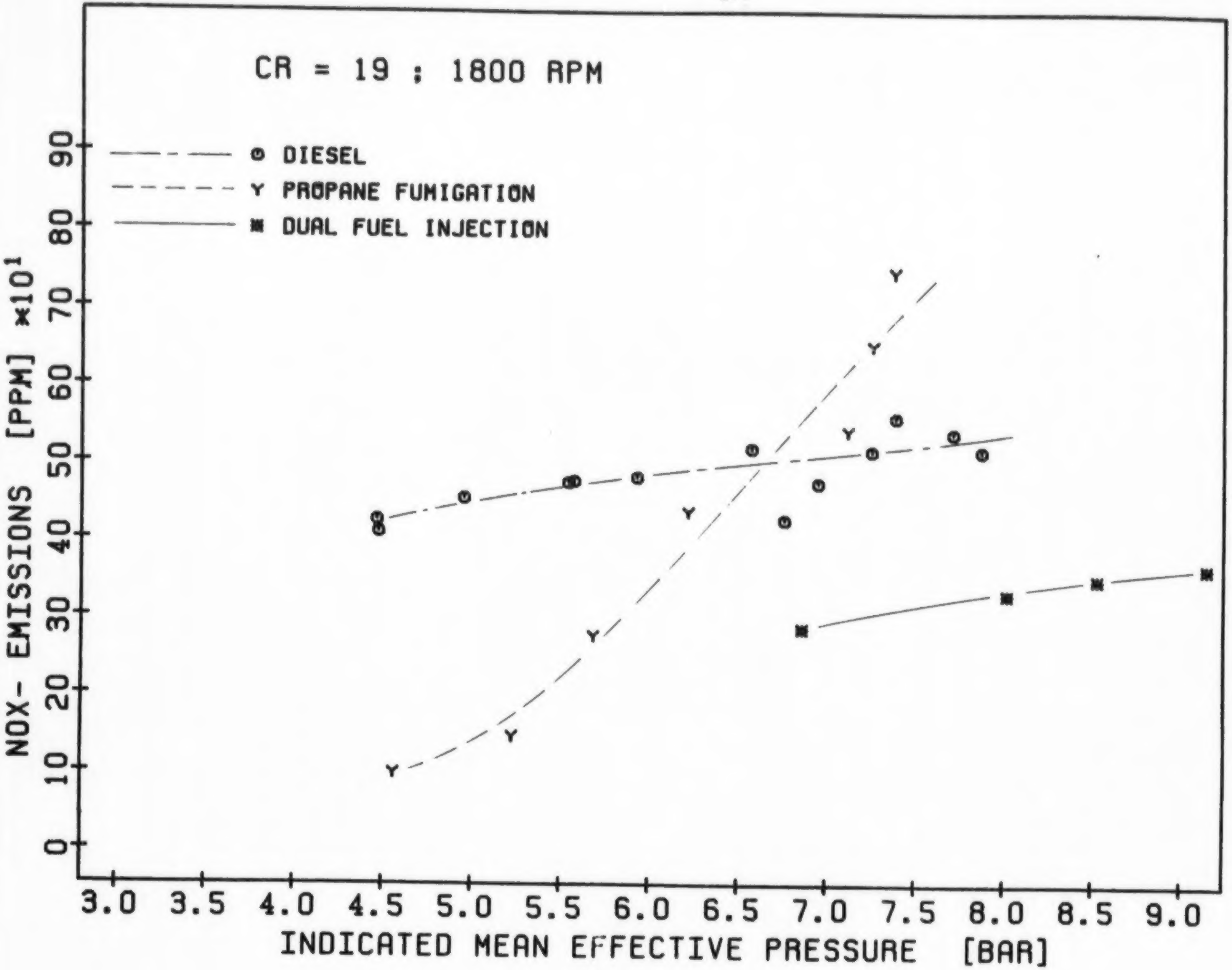

CR = 19 ; 1800 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
NOX- EMISSIONS [PPM] *10¹
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-48

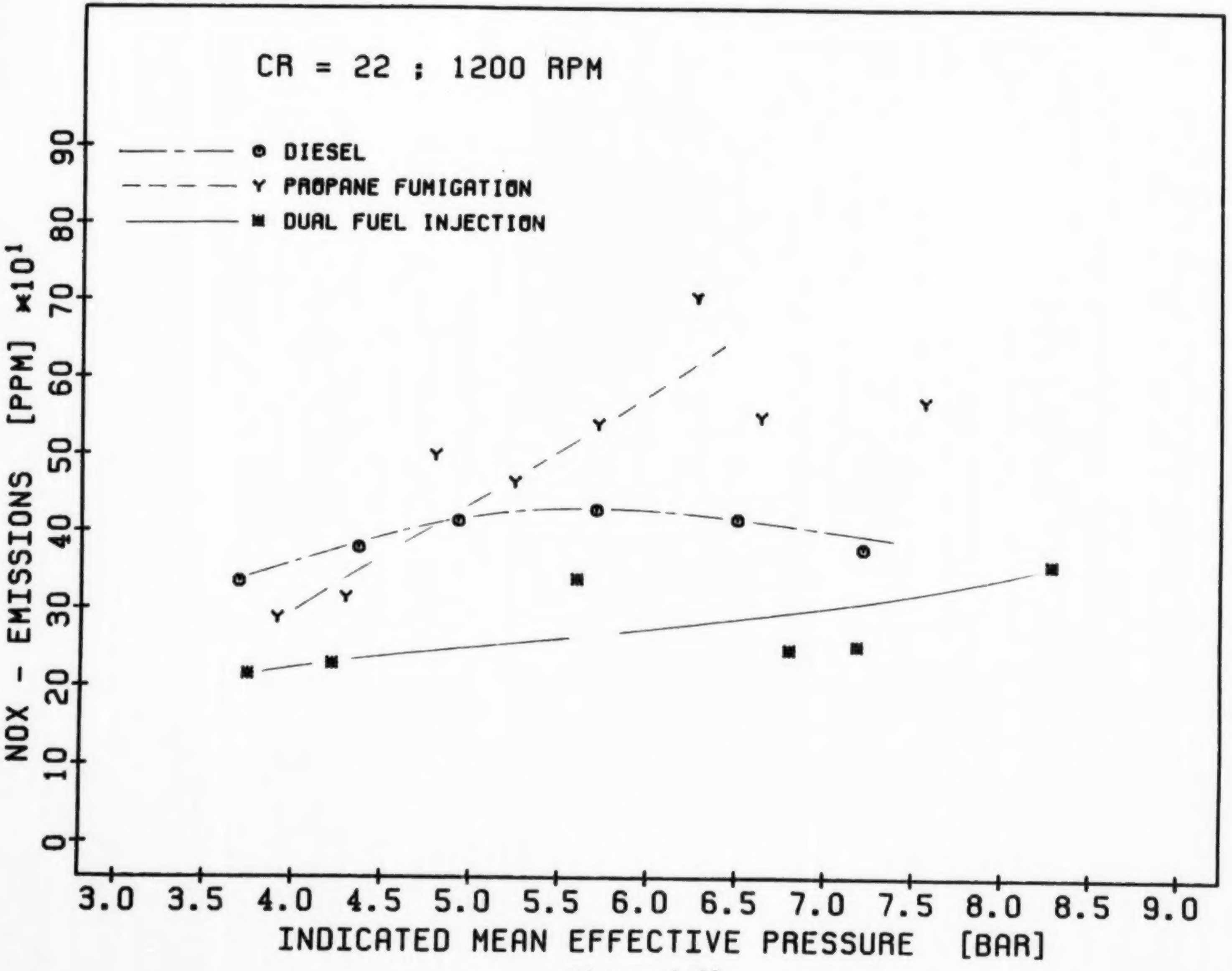

CR = 22 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] ×10¹
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-49

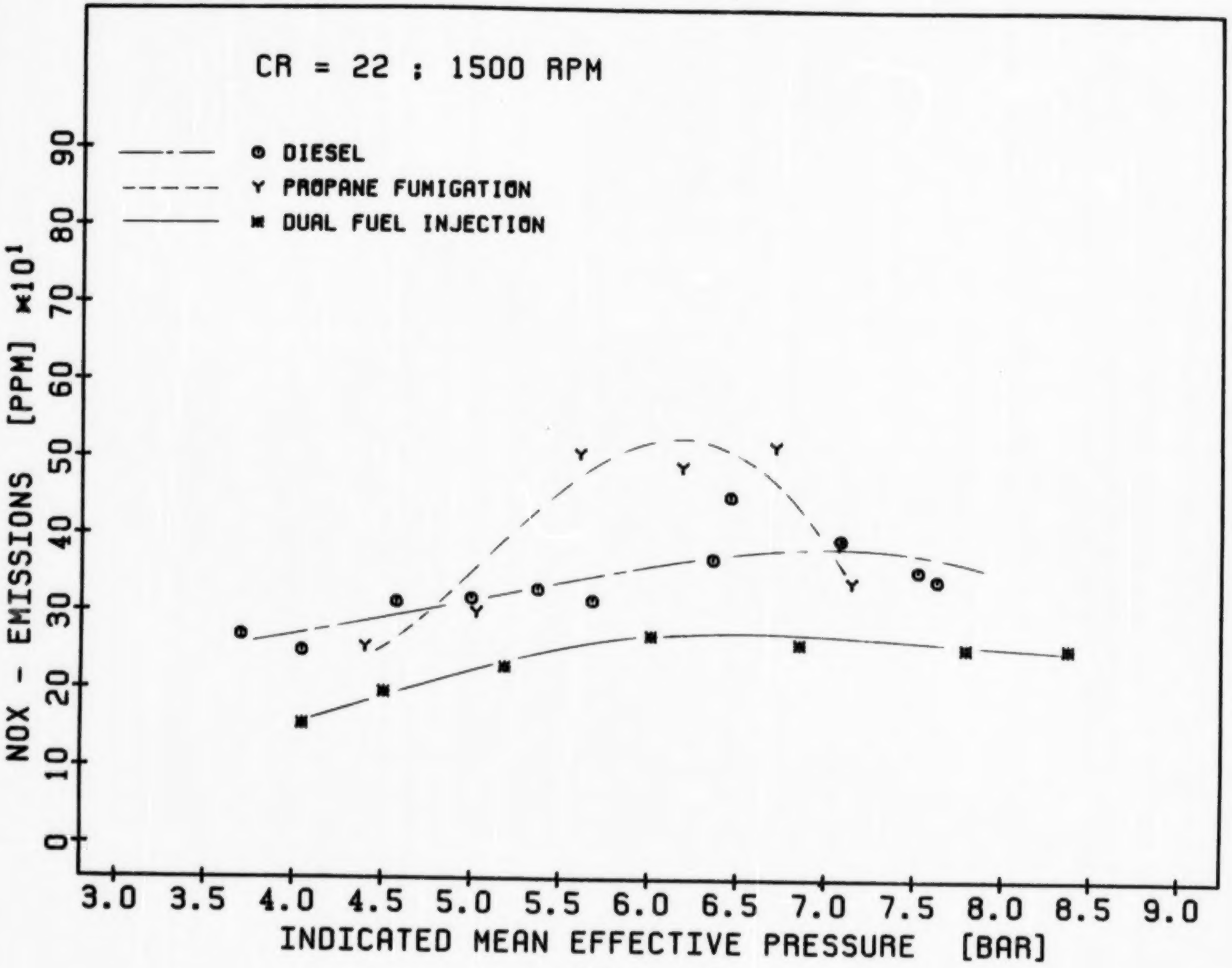

CR = 22 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] ×10¹
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-50

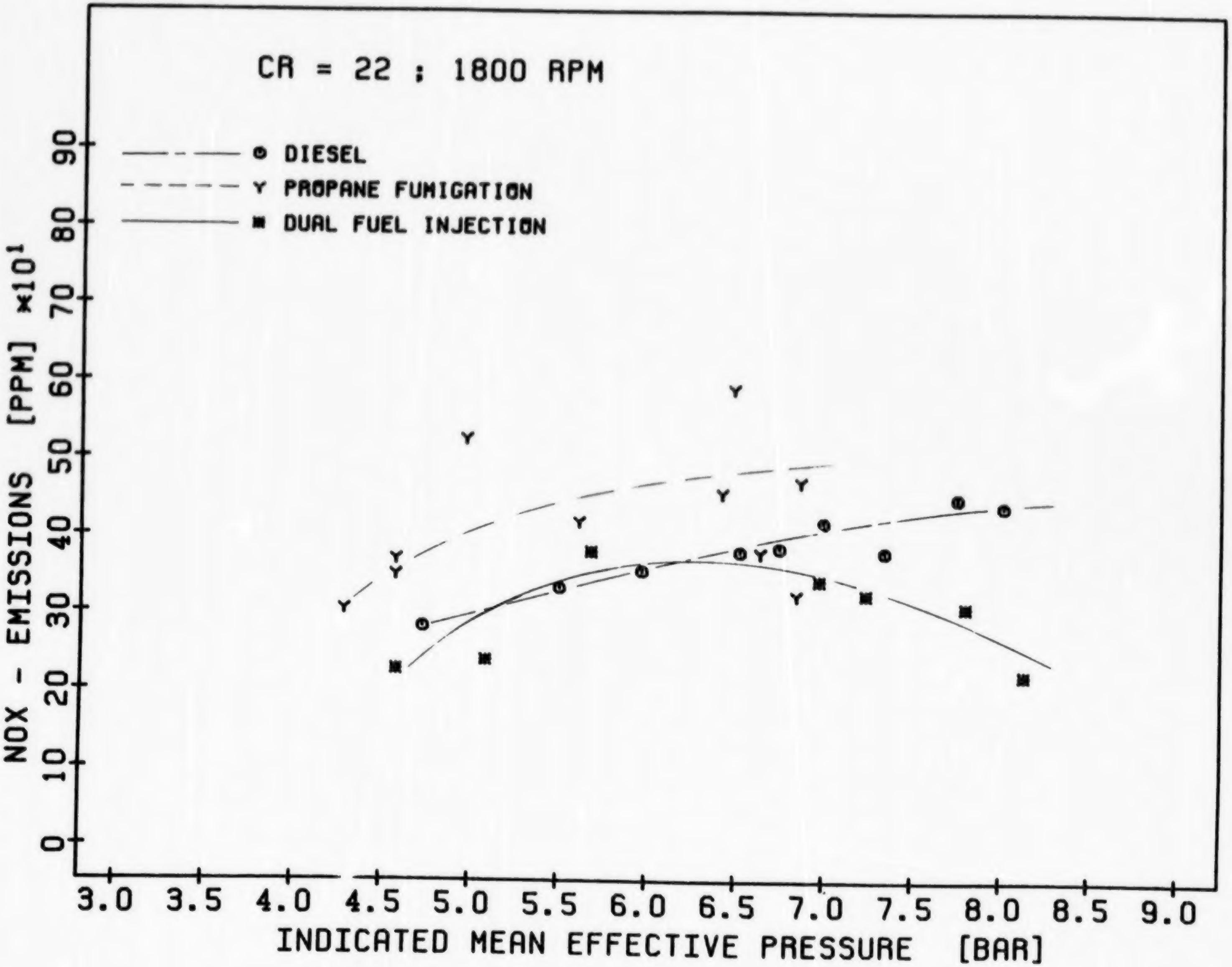

CR = 22 ; 1800 RPM
⊖ DIESEL
Y PROPANE FUMIGATION
■ DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] ×10¹
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-51

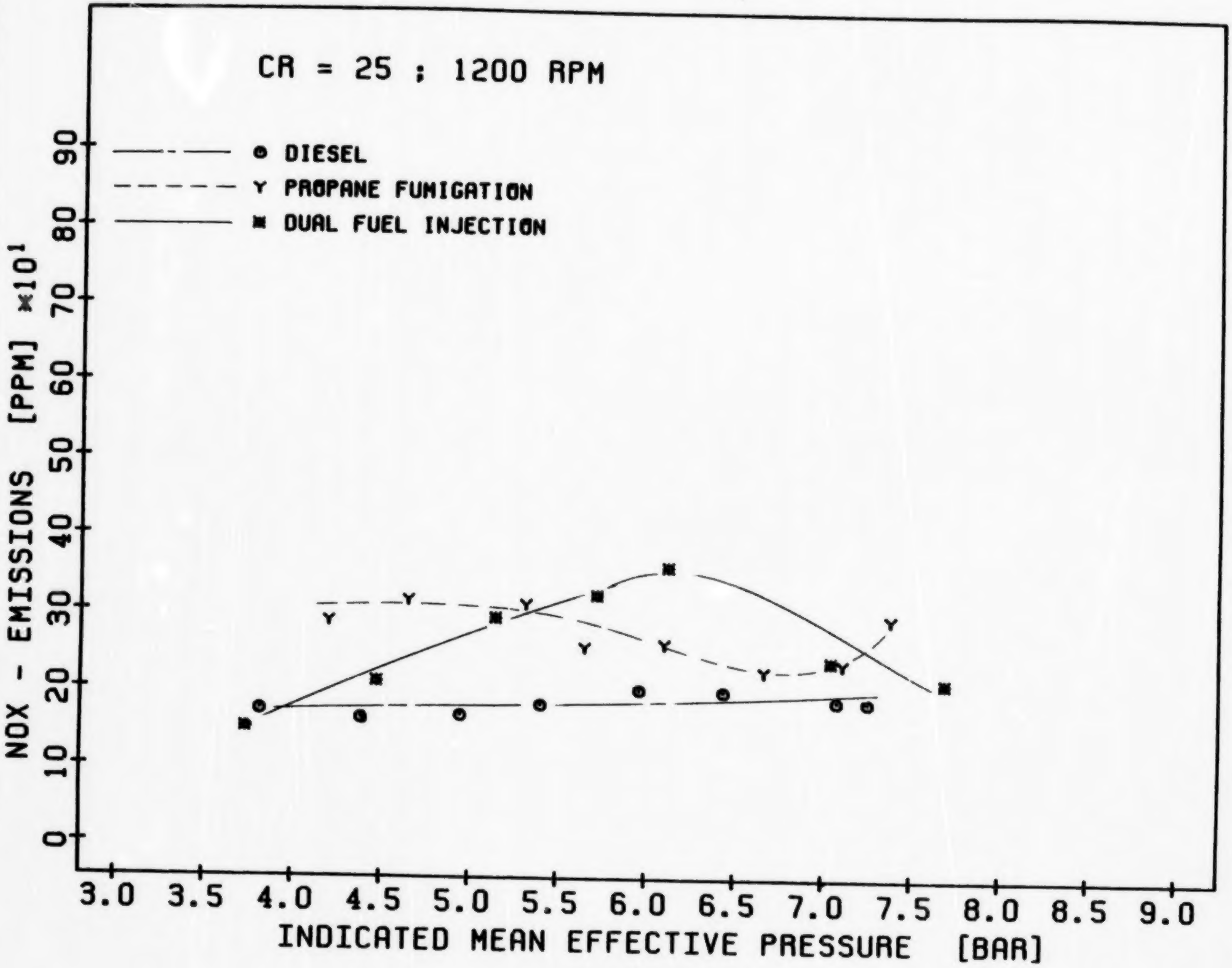

CR = 25 ; 1200 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] ×10¹
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-52

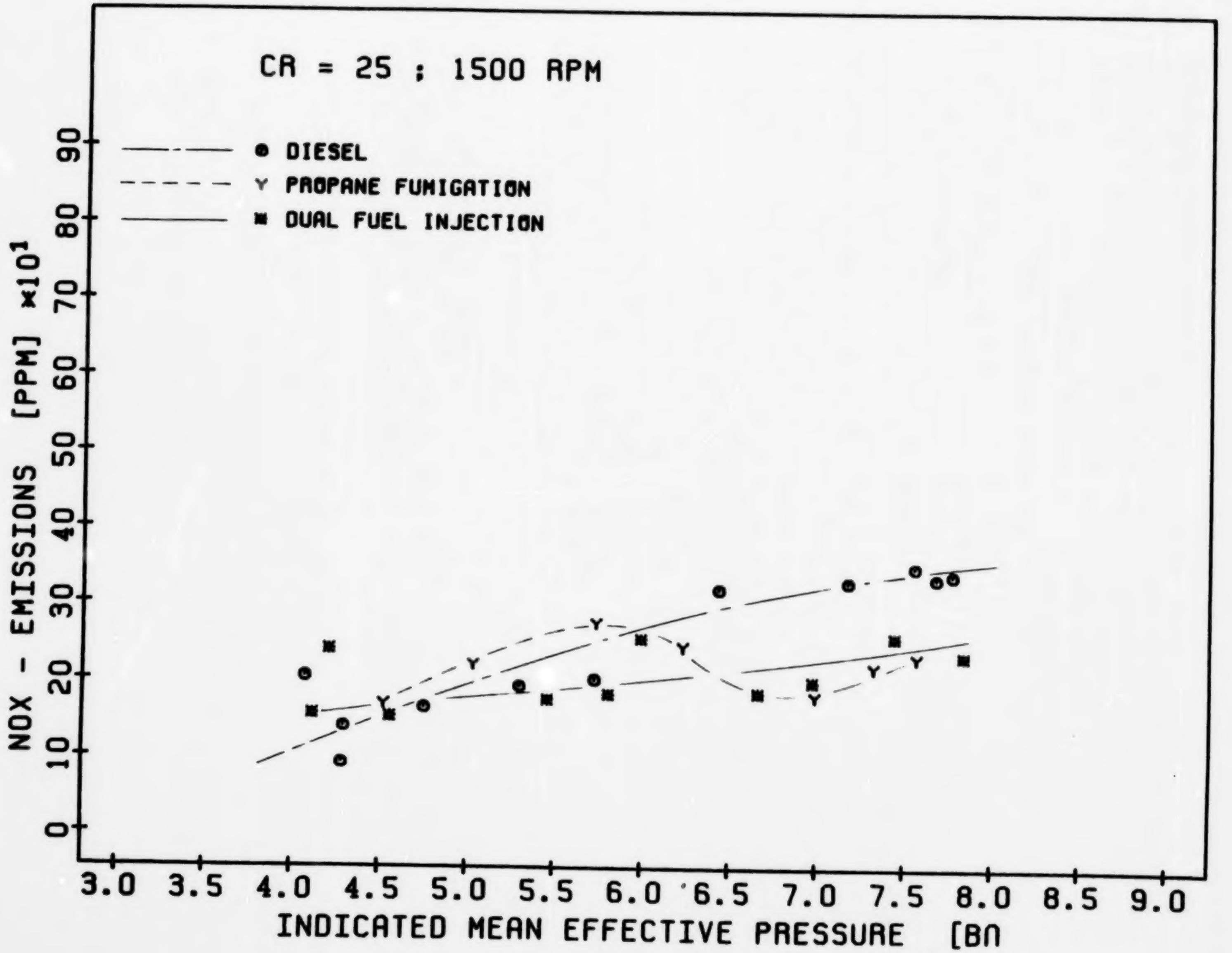

CR = 25 ; 1500 RPM
DIESEL
PROPANE FUMIGATION
DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] ×10^1
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BN

Figure 4-53

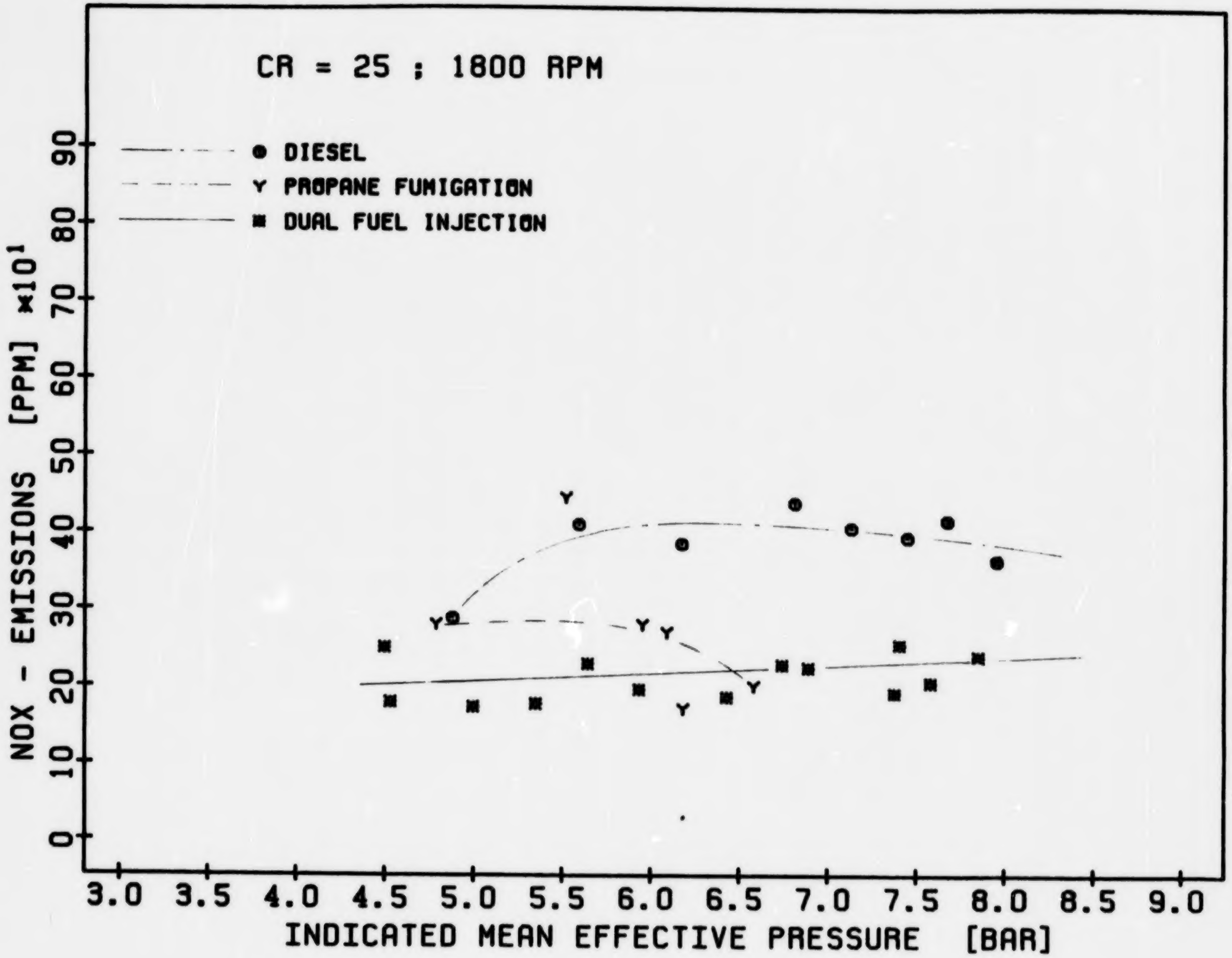

CR = 25 ; 1800 RPM
● DIESEL
Y PROPANE FUMIGATION
■ DUAL FUEL INJECTION
NOX - EMISSIONS [PPM] $\times 10^1$
0
10
20
30
40
50
60
70
80
90
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-54

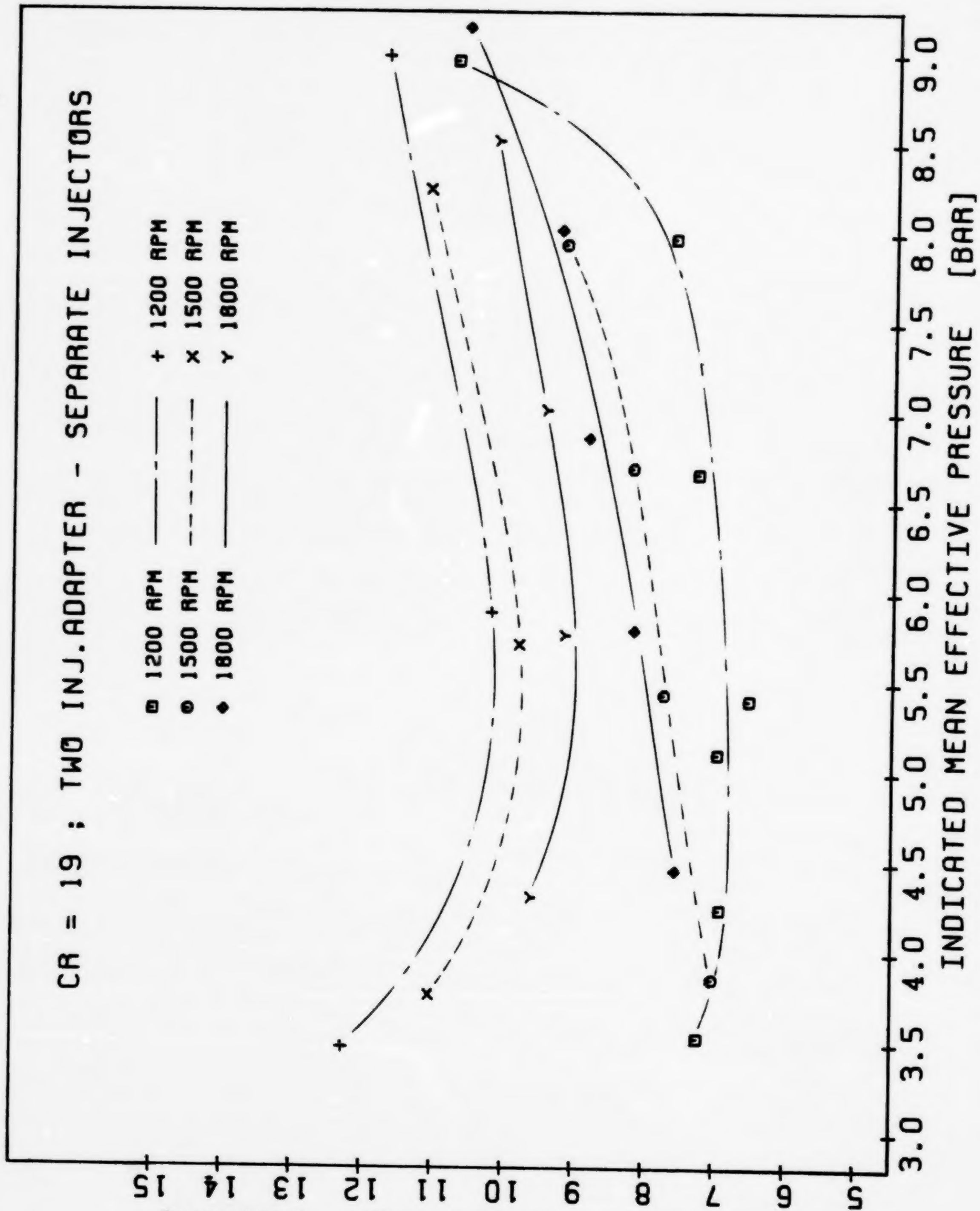

CR = 19 ; TWO INJ.ADAPTER - SEPARATE INJECTORS
⊡ 1200 RPM ——— + 1200 RPM
⊙ 1500 RPM - - - × 1500 RPM
◆ 1800 RPM ——— Y 1800 RPM
INDICATED MEAN EFFECTIVE PRESSURE [BAR]
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
IND.SPEC.FUEL CONSUMPTION [MJ/KWH]
5
6
7
8
9
10
11
12
13
14
15

Figure 4-55

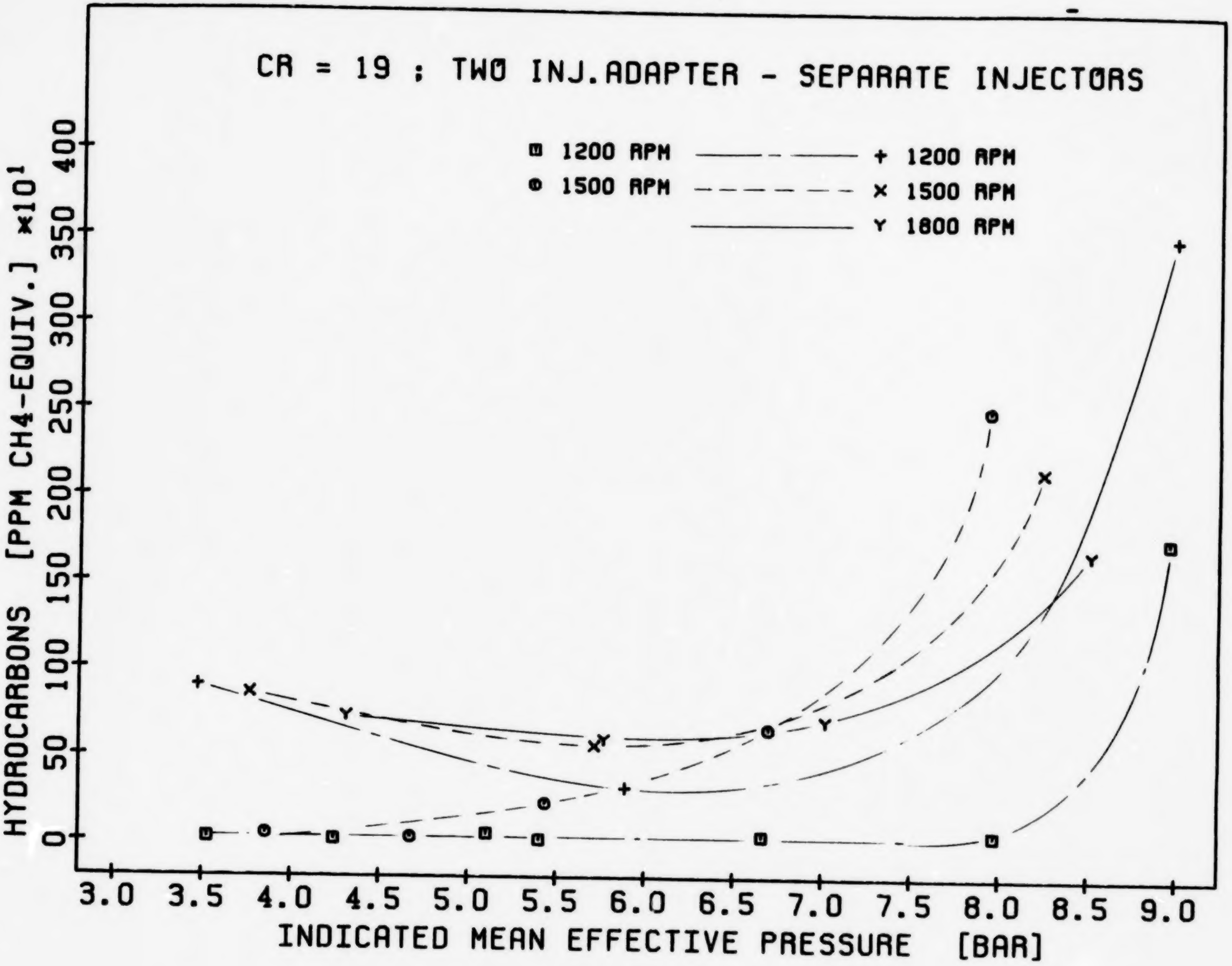

CR = 19 ; TWO INJ.ADAPTER - SEPARATE INJECTORS
□ 1200 RPM
+ 1200 RPM
⊙ 1500 RPM
× 1500 RPM
Y 1800 RPM
HYDROCARBONS [PPM CH4-EQUIV.] ×10¹
0
50
100
150
200
250
300
350
400
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-56

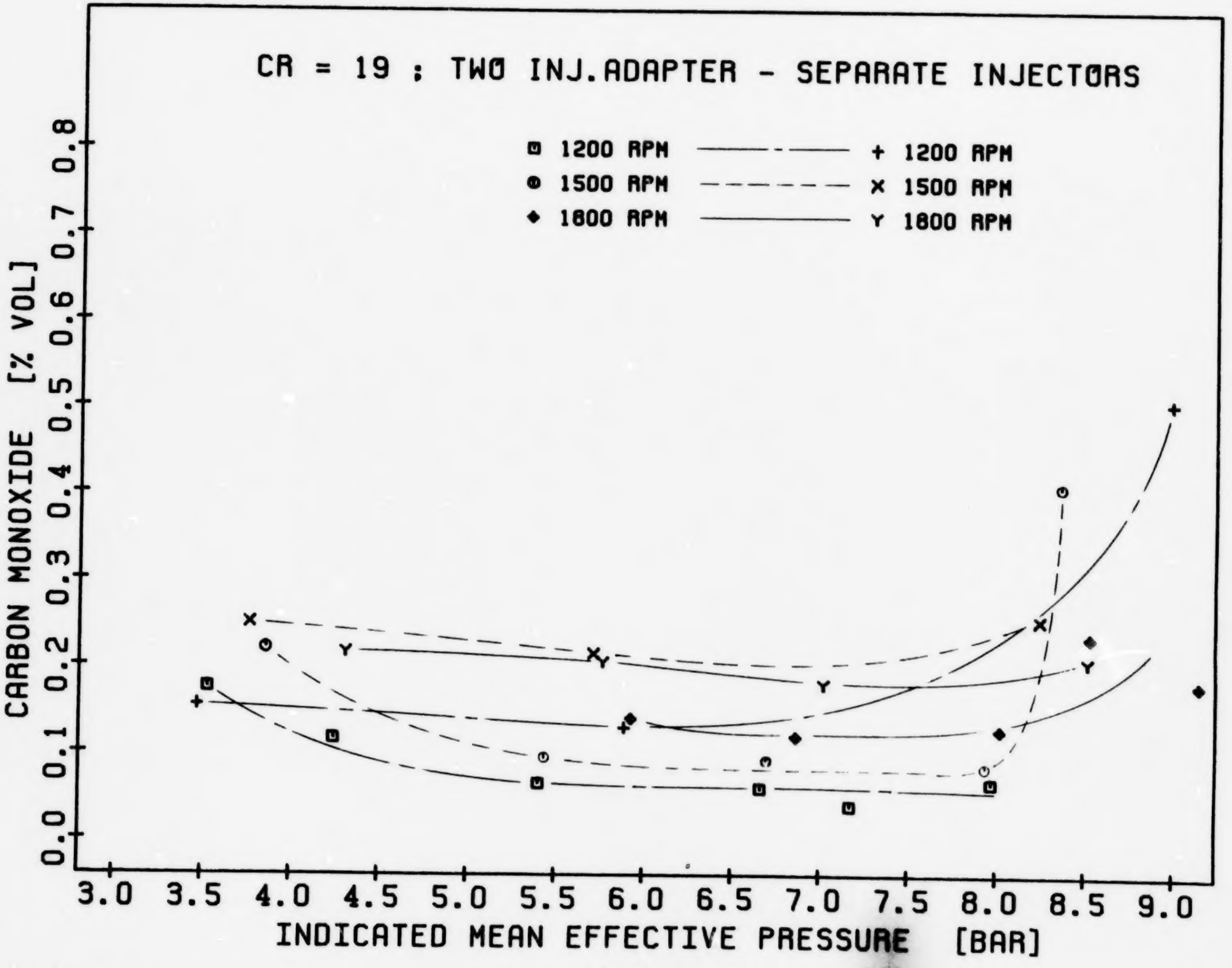

CR = 19 ; TWO INJ.ADAPTER - SEPARATE INJECTORS
□ 1200 RPM
⊙ 1500 RPM
◆ 1800 RPM
+ 1200 RPM
× 1500 RPM
Y 1800 RPM
CARBON MONOXIDE [% VOL]
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-57

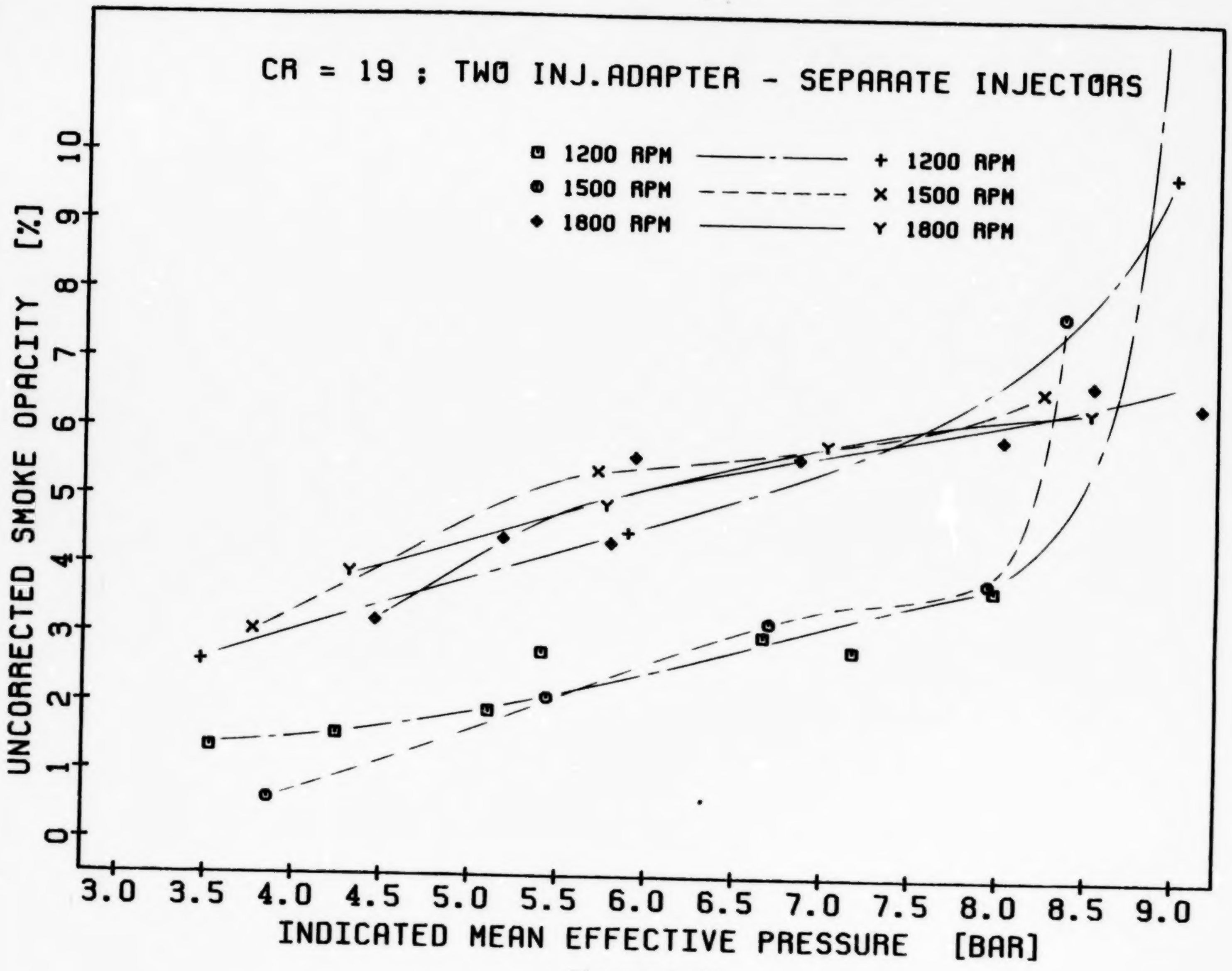

CR = 19 ; TWO INJ.ADAPTER - SEPARATE INJECTORS
□ 1200 RPM —·— + 1200 RPM
● 1500 RPM - - - × 1500 RPM
◆ 1800 RPM —— Y 1800 RPM
UNCORRECTED SMOKE OPACITY [%]
0
1
2
3
4
5
6
7
8
9
10
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-58

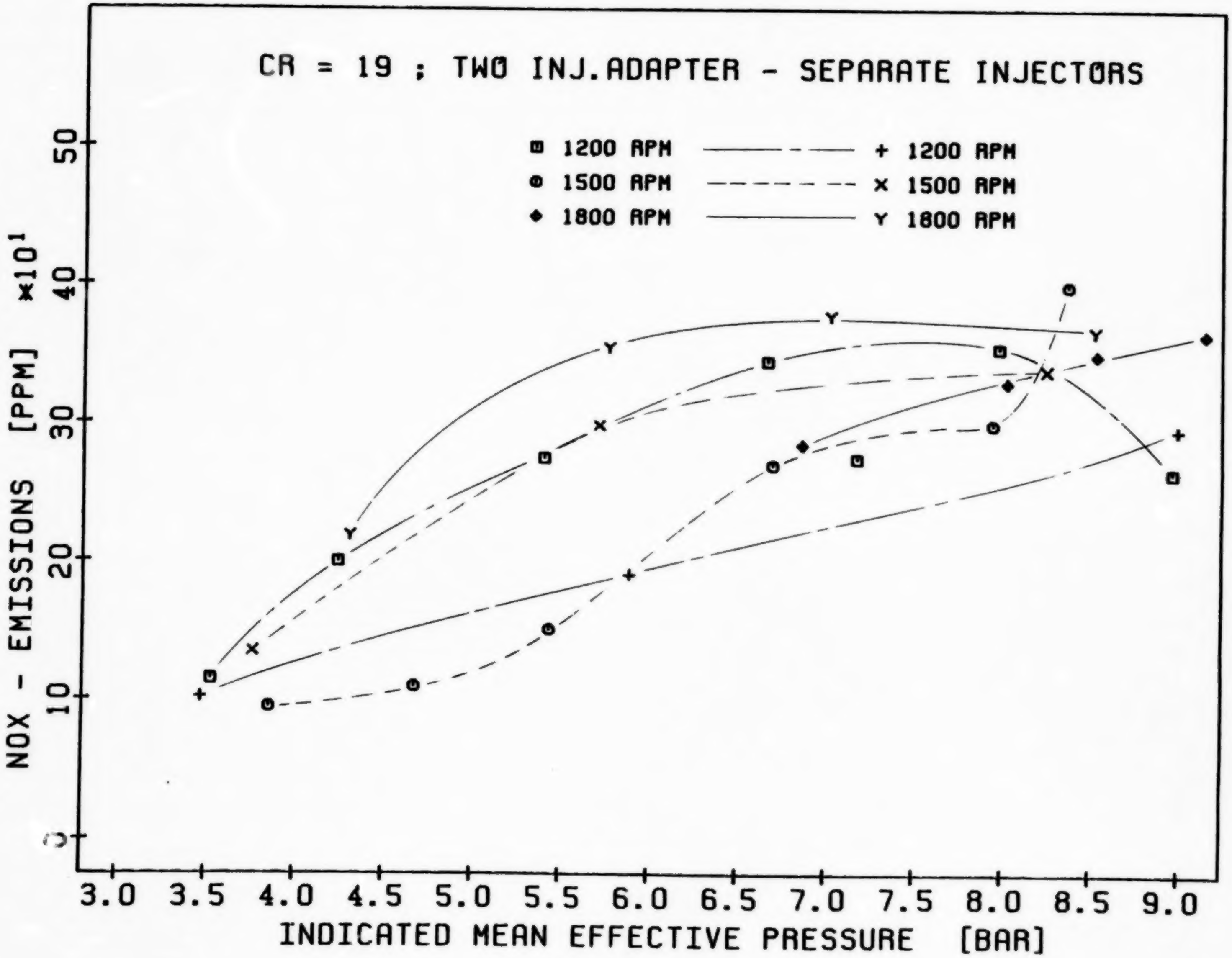

CR = 19 ; TWO INJ.ADAPTER - SEPARATE INJECTORS
1200 RPM
1500 RPM
1800 RPM
+ 1200 RPM
× 1500 RPM
Y 1800 RPM
NOX - EMISSIONS [PPM] ×10¹
0
10
20
30
40
50
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-59

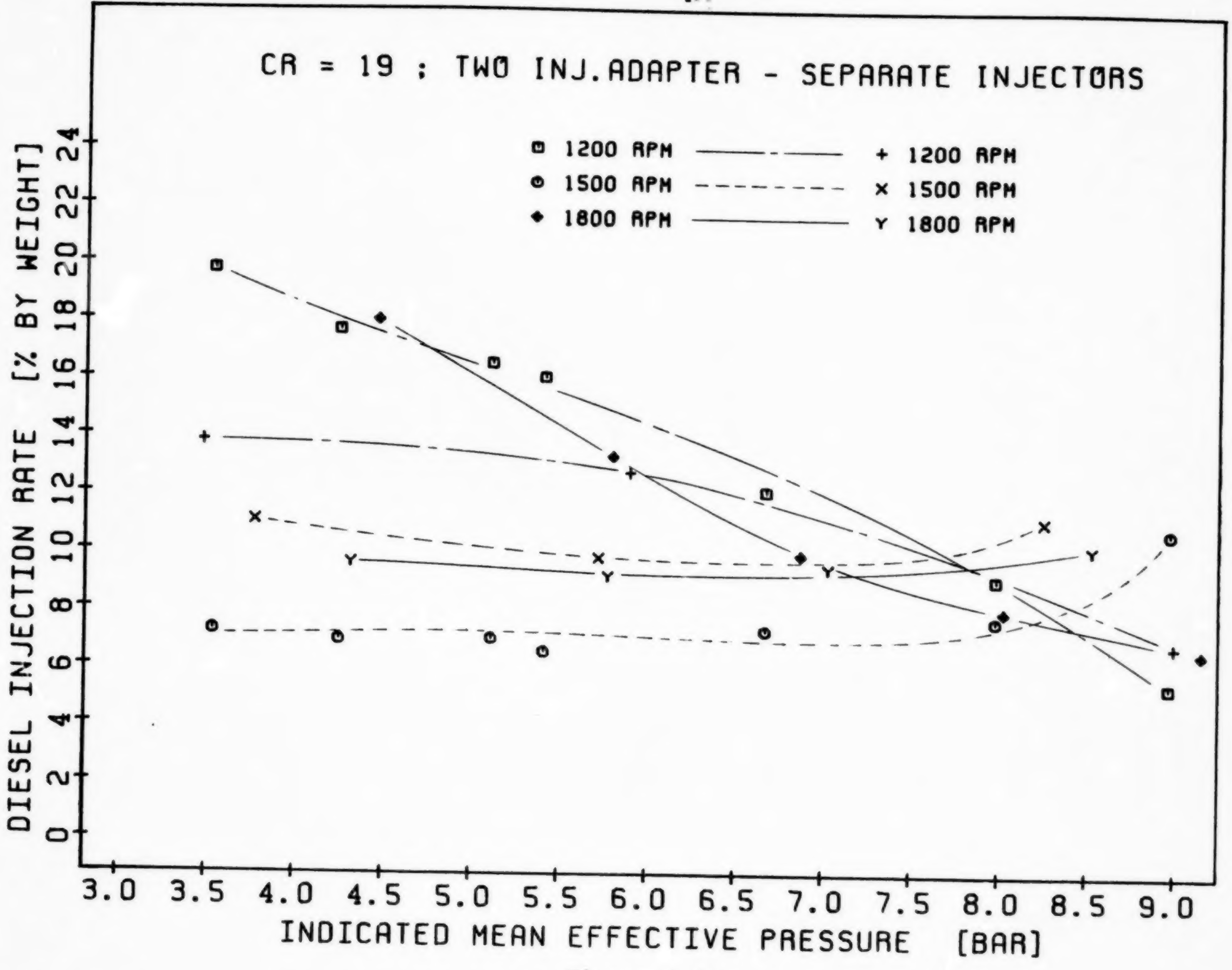

CR = 19 ; TWO INJ.ADAPTER - SEPARATE INJECTORS
1200 RPM
1500 RPM
1800 RPM
+ 1200 RPM
× 1500 RPM
Y 1800 RPM
DIESEL INJECTION RATE [% BY WEIGHT]
0
2
4
6
8
10
12
14
16
18
20
22
24
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
9.0
INDICATED MEAN EFFECTIVE PRESSURE [BAR]

Figure 4-60

5. CONCLUSIONS.

In summary, the use of propane fuel in diesel engines offers the significant advantage of reduced smoke emissions while achieving a power output equivalent to the original diesel. Of the two propane fueling methods examined in this project, dual fueling offers greater propane substitution than can propane fumigation. In addition, the dual fuel engine has lower NO_x emissions throughout the power range and lower CO and hydrocarbon emissions at light loads. At the present state of development, the energy consumption of the dual-fueled engine is comparable to that of the original diesel. Further development effort should produce even better efficiency.

Another important advantage of dual-fueling compared to fumigation is the much greater stability. Engine control was sometimes difficult when fumigation was employed. On the other hand, the dual-fueled engine was stable and easily controllable throughout its entire operating range.

The tradeoff for the advantages of dual-fueling is the increased mechanical complexity of the system. Considerable development is needed to provide a simpler and more reliable engine fueling system. A number of problem areas were identified in this first stage of development which should be addressed in subsequent work.

The first of the prolems encountered is keeping the propane fuel liquified. Heat input to saturated liquid propane creates cavitation problems in the supply pump that propagate through the fuel injection pump and cause erratic injector operation. In the vehicle application, a propane pump must be installed near the tank to increase the propane pressure before the propane encounters high underhood temperatures. This need not be a pressure above the critical pressure if heat conduction from the engine block to the fuel injection pump can be minimized. A high pressure line would be needed to convey the fuel to the engine compartment. Natural gas fueled vehicles already have a much higher pressure fuel supply line running the length of the vehicle.

Wear problems in the fuel injection pump are still not completely solved. Use of alternative materials for wear surfaces may be necessary. Perhaps a coating, e.g. Teflon, would provide the necessary protection. The flow of lubricating oil applied to the pump barrel in this project is not well controlled and could result in significant oil consumption. Furthermore, lubricating oil appears to provide ignition even in the absence of the diesel pilot, but with high smoke emissions. Worth exploring is the concept of mixing a small amount of diesel fuel in with the propane to lubricate the pump and to provide ignition. This could be done at source, i.e. mixed into the propane tank, or adding diesel fuel through the pump oiling groove as was done with lubricating oil in this project. One problem with this idea is that part of the diesel oil would pass down the pump plunger into the lube oil reservoir.

Resonance in the fuel injection line was also a problem and resulted in late secondary injections of propane and subsequent high smoke emissions. This is not a problem peculiar to propane, but requires careful tailoring of the injection system to the particular installation.

Finally, continual difficulty was experienced with plugging of the fine orifices in the injector nozzles.

These problems should be overcome with further development. The results of the first stage of development carried out in this project demonstrate the feasibility and promise of the dual-fuel propane diesel engine.

6. REFERENCES

1. UTME/ERDL Reports for Ontario Ministry of Transportation and Communications. "Propane Fuel/Engine Systems Optimization and Testing", Phases I, II, III, IIIb, IV, and V, 1981-82. Also, Wallace, J.S. and Segal, L., "Consolidation and Assessment of Propane Equipment Test Results", Final Technical Report to Ontario Ministry of Transportation and Communications, Contract No. 31523, March, 1986.

2. Karim, G.A. "A Review of Combustion Processes in the Dual Fuel Engine - The Gas Diesel Engine", *Prog. Energy Combust. Sci.*, *6*, 277-285 (1980).

3. Pischinger, F. "Dieselzweistoffbertrieb mit Flussigeinspritzung von Propane/Butane ins Ansaugrohr", *MTZ Motortech A.*, *40* (3), 121-124 (1979).

4. Oester, U., Wallace, J.S., Segal, L. and Keffer, J.F., "Combustion Characteristics of Propane Directly Injected into Diesel Engines", Final Report to Transportation Development Centre of Transport Canada, DSS Contract File 14SD.T8200-3-3533, Serial No. OSD83-00106., June, 1986.

APPENDIX E

IVI MANUFACTURING COST EVALUATION

IVI Inc.

C.P. 790 COURCELETTE
QUEBEC (Quebec)
G0A 1R0
(418) 844-2711

TLX: 051-2256

A/TO	M. Peter Nikolaїczuk, SNC Inc.	PAGE	
DE/FROM	M. Serge Roy, IVI Inc.	DATE	11-03-86
OBJET/SUBJECT			

COPIE A/TO

Peter,

Tel que demandé, tu trouveras ci-inclus le document concernant les coûts de production d'un double système d'injection.

Espérant le tout à ton entière satisfaction, n'hésite pas à communiquer avec moi si tu as besoin de plus amples renseignements.

Serge Roy

Serge Roy, ing.
Dépt. Recherches & Développement

SR/rb

p.j.

INTRODUCTION

La présente a pour objet de définir grossièrement les coûts de production d'un double système d'injection. Deux options ont été étudiées.

La première implique l'utilisation d'injecteurs standards fabriqués par Stanadyne montés dans un support. C'est la fabrication de ce support et le montage des injecteurs dans celui-ci qui sont brièvement étudiés.

La seconde option est une fabrication complète de l'injecteur. Le support est usiné de manière à reproduire fidèlement l'intérieur d'un injecteur standard. Toutes les pièces à l'exception de quelques-unes standards sur le marché sont étudiées en fonction de leur coût de fabrication. L'étude porte sur une production de petite, moyenne et de grande envergure. Pour chacune des productions, on y définit les étapes de réalisation de chacune des pièces du système et un temps de fabrication y est associé. Les équipements nécessaires à la fabrication y sont sommairement décrites ainsi que leur coût d'achat respectif. Le tout est résumé dans les Tableaux I à IV.

HYPOTHESES DE BASE

Cette étude se voulant une première ébauche des coûts de production d'un double système d'injection propane-diesel pour de petites et grandes productions, il est nécessaire de définir les hypothèses de base considérées dans celle-ci.

- Aucun temps n'est associé pour le développement du prototype du système. Celui-ci est considéré comme fonctionnel et répondant aux aspirations de ses concepteurs. L'option 1 ayant déjà un prototype de construit et de vérifier n'a en principe aucun autre développement à considérer. Cependant, l'option 2 n'a jamais été vérifiée et n'a à son actif aucun développement de fait. L'étude se veut entièrement théorique et est basé sur les faibles informations fournies par Stanadyne sur la fabrication de leurs injecteurs.

- Les équipements de production ont été évalués au meilleur de notre connaissance et leurs coûts ont été approximés en fonction des équipements connus à ce jour. Les équipements spécialisés hors série nécessitant une étude très approfondie pour en définir leur taux de production et leurs coûts ont aussi été approximés de façon superficielle.

- Les coûts de la main d'eouvre ont été évalués à $40./heure. Ceux-ci incluent les frais généraux, la main d'oeuvre directe, et l'administration générale. Aucun profit n'est pris en considération. Il en est de même pour l'amortissement des équipements. Ce taux horaire est jugé être le taux au coûtant.

- Les coûts de production sont évalués pour des bases de production de 1, 10, 10,000 et 100,000 unités.

- Les frais de manutention et d'essais sont évalués à 10% des coûts de fabrication.

OPTION 1 - SUPPORT CONTENANT DEUX INJECTEURS STANADYNE

Cette option fait appel à l'utilisation de deux injecteurs Stanadyne montés à angle dans un support tel qu'illustré à la Figure 1.0. Les injecteurs sont respectivement utilisés pour l'injection du propane et du diesel. Le diamètre des ouvertures des embouts des injecteurs sont de 0.005 pouce pour le diesel (1) et 0.011 pouce pour le propane (2). Ceux-ci ne sont pas standards et doivent être fabriqués sur commande par Stanadyne.

Une autre solution est d'utiliser des injecteurs standards et d'y boucher les trous excédentaires à l'aide de différents produits tel le Loctite. Cette dernière n'est pas recommandée dû aux grandes difficultés à boucher efficacement les ouvertures.

Pour les raisons d'encombrements physiques, le bout de chacun des injecteurs est usiné de maniere à y faire un méplat (Figure 1.2). Une fois monté dans le support, les deux injecteurs sont retenus ensemble par une époxie que l'on injecte sous pression entre les méplats. Un espace de 0.001 pouce est laissé entre les méplats pour permettre à l'époxie de bien couvrir les surfaces à unir et ainsi obtenir un meilleur mariage des deux pièces. Une graisse doit être appliquée au bout des injecteurs pour empêcher que l'époxie ou tout autre matière bouchent les trous des embouts.

La fabrication du support d'injecteur est la partie la plus difficile à faire (Figure 1.1). Elle se fait en quatre étapes. Premièrement, le dégrossissage et le tournage des diamètres extérieurs se fait sans trop de difficultés. Cependant, le perçage des trous d'insertion des injecteurs demande beaucoup plus d'attention. Les trous sont percés à angle de 2° avec le centre sur une longueur de 3 pouces. Une fois le premier trou percé, celui-ci est bouché pour empêcher la mèche de dévier lors du perçage du second trou. L'intérieur de ceux-ci est ensuite rectifié pour obtenir le fini de surface et la précision voulue.

Finalement, une fois les injecteurs montés dans le support, le diamètre de 0.549 pouce est tourné avec précaution pour ne pas endommager les embouts.

Production - 1 pièce

Pour les étapes de tournage et de dégrossissage des diamètres extérieurs du support d'injecteur, un tour conventionnel de bonne précision fera l'affaire. Les trous seront percés à l'aide d'une fraiseuse standard montée avec une tête à diviser. Un montage spécial doit être prévu pour tenir la pièce à angle. Dû à la haute précision du diamètre et à la rectitude des trous, ceux-ci seront rectifiés à l'aide d'une rectifieuse montée sur le tour.

Le montage des injecteurs dans le support est de type dégagé et ne nécessite aucune force d'insertion. L'injection de l'époxie entre les méplats des deux injecteurs se fera à l'aide d'une pompe manuelle d'injection.

Finalement, une fois les injecteurs scellés dans l'époxie, l'ensemble support-injecteurs est monté sur le tour pour dégrossir le diamètre de 0.549 pouce. Les coupes de finition sont faites à la rectifieuse cylindrique pour ne pas endommager le bout des injecteurs.

Production - 10 pièces

Pour une si faible quantité, il n'est pas nécessaire d'utiliser des appareils à contrôles numériques. Le même équipement que celui utilisé à l'étape précédente sera suffisant. Cependant, l'expertise acquise pour la fabrication du premier injecteur permet de diminuer le temps de réalisation des autres de l'ordre de 50%.

Production - 10,000 pièces

Pour une telle production, il est nécessaire d'utiliser des machines à contrôle numérique. Un tour, et une fraiseuse à contrôle numérique permettront de réduire considérablement le temps d'usinage. Un centre d'usinage ne serait pas tellement utile car le perçage des trous ne nécessire pas un changement fréquent des outils.

Les pièces pourront être rectifiées sur une rectifieuse cylindrique à commande numérique.

Pour injecter l'époxie, on pourra faire appel à un système à multiples points d'injection réduisant ainsi la durée de l'opération.

Le temps d'usinage d'une pièce sera limité par le temps de perçage des trous d'insertion des injecteurs. L'augmentation du nombre d'équipements de perçage contribuerait grandement à accélérer la production. Toutefois, la présente étude ne tient compte que d'une seule ligne de production.

Production - 100,000 pièces

Une telle production exige de se tourner vers des appareils automatisés dédiés spécifiquement à la tâche à accomplir. Ces appareils doivent en partie être développés par des compagnies ayant l'expertise dans le domaine. Ces appareils peuvent augmenter la capacité de productin par un facteur estimé de l'ordre de 3. La fraiseuse serait remplacée par un tel équipement permettant ainsi de percer conjointement les deux ouvertures.

Un tour à chargement rotatif à multi-têtes d'usinage pourrait être utilisé pour le dégrossissage, le tournage et le meulage de la pièce.

De même que dans les productions de moindre envergure, un appareil à multiples points d'injection d'époxie pourrait être utilisé pour faire cette opération. Le tout pourrait être automatisé.

OPTION 2 - FABRICATION COMPLETE DU SUPPORT ET DES INJECTEURS

Dans l'optique de concurrencer un jour Stanadyne ou bien de diminuer nos coûts de fabrication du système, la fabrication en entier du double système à injection est une solution à envisager. Pour ce faire, nous pouvons soit acheter la technologie de fabrication des injecteurs de Stanadyne ou bien de développer notre propre expertise dans le domaine. Cette dernière nécessitant plusieurs années de recherche et de développement est difficilement envisageable. Voilà pourquoi cette option est basée entièrement sur le concept de Stanadyne. Tel qu'illustré à la Figure 2.0, le support est usiné de manière à représenter fidèlement l'intérieur d'un injecteur standard Stanadyne. Toutes les pièces sont fabriquées en conformité avec les dimensions des pièces standards. Les étapes de fabrication de chacune de ces pièces sont résumées dans le Tableau III. On y retrouve également une approximation du temps de fabrication. L'équipement nécessaire est donné au Tableau IV.

Corps de l'injecteur

Le corps de l'injecteur (Figure 2.1) est la partie la plus complexe à usiner dû aux faibles diamètres à percer, à la longueur de la pièce, l'angle de perçage et la précision demandée. La pièce est premièrement dégrossie pour y tourner les diamètre extérieurs. Seul le diamètre de 0.549 pouce est tourné en dernier lieu, une fois les injecteurs complètement assemblés.

Les trous d'insertion des mécanismes sont ensuite percés. Premièrement, les diamètres du haut de l'injecteur sont percés pour servir de guide à la mèche de plus petit diamètre. Ce dernier est percé en plusieurs étapes avec des mèches de différentes longueurs pour éviter le risque de désenlignement du trou. La pièce est ensuite retournée pour percer les trous d'insertion des embouts. Celui-ci est ensuite rectifié pour assurer la concentricité des diamètres du haut et du bas du support.

Finalement, la pièce est retrounée à 90° pour percer les trous d'insertion des conduites d'alimentation en combustible.

Embout

L'embout (Figure 2.2) est fabriqué à partir de tiges d'acier. Les diamètres extérieurs sont en premier lieu tournés. La cavité est ensuite percée. La pièce doit subir un traitement thermique de l'ordre de 55 à 60 Rc pour résister à l'usure et l'érosion du fluide sous pression. L'extérieur de l'embout ainsi que l'intérieur du sac où siège la valve sont finis à la meule à très haute précision. Finalement, les trous d'évacuation du combustible sont percées à l'aide d'une appareil d'usinage par électro-érosion (EDM) ou bien de perceuse hautement sophistiquée. Les diamètres de ces trous (2) sont de 0.011 pouce pour le propane et de 0.005 pouce (1) pour le diesel.

Valve

La valve telle qu'illustrée à la Figure 2.3 est une pièce très importante pour le bon fonctionnement du système. Un soin particulier doit être porté à l'usinage du diamètre de la tête de la valve. Une précision de 0.0001 pouce y est nécessaire.

Après tournage des diamètres extérieurs, la pièce doit subir un traitement thermique de 55 à 60 Rc. Les pièces sont trempées verticalement pour diminuer le fléchissement de celles-ci. Finalement, elles sont meulées pour obtenir le fini de surface et la précision voulue.

Guide de valve

L'étape importante dans la fabrication du guide de valve (Figure 2.4) est d'obtenir un excellent fini de surface et une tolérance très serrée du diamètre du trou. Pour ce faire, une fois les étapes de tournage et filetage terminées, une balle de diamètre très serré est forcée dans cette ouverture pour obtenir une ouverture parfaitement cylindrique. Ce procédé est appelé "ballizing". Celui-ci subit finalement un traitement thermique à atmosphère contrôlée pour obtenir une dureté de 55 à 60 Rc.

Siège du ressort

Le siège du ressort (Figure 2.5) est une pièce très simple à fabriquer et ne nécessite qu'un tournage et un traitement thermique.

Vis d'ajustement de la pression et d'ajustement de l'ouverture

Ces deux pièces sont facilement réalisables et ne nécessitent rien de particulier (Figures 2.6 et 2.7)

Ressort, cale, écrous

Toutes ces pièces sont standards sur le marché.

Conduite d'alimentation de carburant

La conduite d'alimentation de carburant (Figure 2.8) est fabriquée à partir d'un tuyau d'acier sans joint. L'une des extrémités est tournée au diamètre de jonction avec l'injecteur. A l'autre extrémité, un trou de plus grand diamètre est percé pour y insérer le filtreur et le bout est évasé de manière à améliorer l'étanchéité entre les conduites.

Filtreur et connecteur

Ces deux pièces sont facilement réalisables et ne nécessitent rien de particulier (Figure 2.9).

Assemblage des pièces

Le conduit d'alimentation est assemblé au corps par une brazure au four. Il en est de même pour l'embout de l'injecteur. Ces étapes peuvent être faites en même temps.

Le guide de la valve est vissé au corps et un joint d'époxie en assure l'étanchéité. Pour ce faire, un joint de caoutchouc est installé à l'extrémité des filets adjacents à une faible quantité d'époxie. En vissant le guide dans le corps, le joint de caoutchouc est entraîné entre les filets et l'époxie est ainsi aspiré entre ceux-ci. Avant que l'époxie ne durcisse, une tige d'alignement est insérée dans le guide et dans l'embout pour aligner parfaitement les deux pièces. L'époxie durcissant, le guide se retrouve fixé et aligné.

Finalement, les autres pièces sont installées dans le corps et le tout est vérifié sous pression pour détecter les fuites.

Production - 1 pièce

Toutes les opérations de tournage et de dégrossissage des pièces peuvent être faites sur un tour conventionnel de très bonne précision. Le meulage et le rectifiage peuvent être faits avec une rectifieuse cylindrique montée à même le tour.

Le perçage des cavités doit être fait sur une fraiseuse à contrôle numérique munie d'une tête à diviser très rigide. Un montage spécial doit être prévu pour maintenir le corps à angle lors du perçage et du rectifiage. Le temps de réalisation de cette étape est très considérable compte tenu de la précision du montage pour l'alignement des cavités.

Un appareil d'usinage par électro-érosion (EDM) ou une perçeuse très spécialisée est indispensable pour le perçage des ouvertures des embouts.

Un four conventionnel peut être utilisé pour le traitement thermique et la brazure.

Un appareil spécial pour faire le "ballizing" du guide de valve doit être fabriqué pour la cause.

Une presse mécanique à faible tonnage peut servir à faire l'évasement au bout de la conduite d'alimentation. Celle-ci peut ensuite être pliée sur une cintreuse manuelle.

Un montage spécial doit être prévu pour l'alignement du guide de valve avec l'embout.

Production - 10 pièces

Pour une si faible quantité, il n'est pas indispensable d'utiliser les appareils à commandes numériques. Le même équipement que celui utilisé à l'étape précédente sera suffisant. Cependant, l'expertise acquise pour la fabrication du premier injecteur permet de diminuer le temps de réalisation des autres de l'ordre de 50%.

Production - 10,000 pièces

Pour une telle quantité, il est nécessaire d'utiliser au minimum des appareils à contrôle numérique. Un tour à contrôle numérique sera indispensable. Cependant, la fraiseuse se doit d'ètre remplacée par un centre d'usinage automatisé. Celui-ci permet le changement des outils automatiquement. C'est l'opération critique qui détermine le taux de production.

Le meulage et le rectifiage se doivent d'être faits sur une restifieuse cylindrique à commande numérique.

Les traitements thermiques peuvent être faits dans une fournaise conventionnelle à grand volume. Plusieurs pieces à la fois peuvent y être traitées.

Le système EDM reste le même que celui à faible production. Un appareil à multi-poinçons se doit d'être développé pour effectuer le "ballizing".

La presse et la cintreuse mécanique peuvent être utilisées pour former l'évasement au bout des conduites d'alimentation ainsi que pour donner la forme finale.

L'opération critique d'alignement du guide de valve avec l'embout doit toujours se faire manuellement avec un montage spécial.

Pour accélérer le taux de production, il faut soit accélérer l'opération de perçage des ouvertures du corps ou augmenter le nombre de centres d'usinage. Cette dernière solution s'avère très dispendieuse.

Production - 100,000 pièces

Avec une production de cette envergure, nous devons nous tourner vers des appareils automatisés dédiés spécifiquement pour l'opération à accomplir. Ces appareils doivent en partie être développés par des compagnies ayant l'expertise dans le domaine. A titre d'exemple, un appareil pour percer les ouvertures dans le corps de l'injecteur serait muni de deux têtes de perçage placées naturellement à angle. Des outils de forme seraient utilisés pour diminuer le nombre de passes. Ces appareils pourraient augmenter la production par un facteur estimé de l'ordre de 3.

Le tournage et le filetage des pièces pourrait être fait sur un tour à chargement rotatif (6 spindles) à multi-têtes d'usinage. Le même genre d'appareil pourrait être utilisé pour le meulage.

Une ligne automatisée pour le traitement thermique devrait être conçue ainsi que pour la brazure au four.

Le système EDM serait à multi-têtes. Les opérations d'avancement et de cintrage de la conduite d'alimentation peuvent être faites automatiquement sur une presse conventionnelle à matrices multiples.

Un appareil spécial pour aligner automatiquement le guide de valve avec l'embout doit être développé.

TABLEAU I - Description des composantes de l'option 1

COMPOSANTES ET ETAPES DE FABRICATION	TEMPS DE REALISATION POUR LES PRODUCTIONS SUIVANTES (minutes/pièce)			
	1	10	10,000	100,000
1. Support d'injecteurs (Figure 1.1)				
- Dégrossissage & tournage des dia. extérieurs	120	60	10	3
- Perçage des trous d'insertion des injecteurs	660	330	45	12
- Montage des injecteurs avec époxie	120	60	10	3
- Tournage du diamètres du bout du support (dia. 0.550)	20	10	3	1
2. Injecteurs (Figure 1.2)				
- Tournage du diamètre extérieur (dia. 0.311)	20	10	3	1
- Usinage des méplats sur les embouts	25	13	3	1
A - TOTAL:	965	483	74	21
- Manutention et essais: 10% de A	96.5	48.3	7.4	2.1
- Imprévus: 10% de A	96.5	48.3	7.4	2.1
B - TOTAL:	1158	579.6	88.8	25.2

TABLEAU I (suite)

COMPOSANTES ET ETAPES DE FABRICATION	TEMPS DE REALISATION POUR LES PRODUCTIONS SUIVANTES (minutes/pièce)			
	1	10	10,000	100,000
- Coût de main d'oeuvre: $40. X B/60	772	386.4	59.2	16.8
- Matériel: - 2 injecteurs Stanadyne - autres: acier, époxie, etc.	400 20	400 20	300 5	300 5
Coût total unitaire:	1192 =====	806.4 ======	364.2 =====	338.60 ======
Taux de production d'une ligne: pièce/heure	0.05 ====	0.1 ====	1.34 ====	5 ===

TABLEAU II - Coût des équipements prévus pour l'option 1

DESCRIPTION	COUT DES EQUIPEMENTS ($K)			
	1	10	10,000	100,000
- Tour standard de bonne précision muni d'une rectifieuse	30	30	--	--
- Tour à contrôle numérique	--	--	150	--
- Fraiseuse à affichage numérique muni d'une tête à diviser	25	25	--	--
- Rectifieuse cylindrique à commande numérique	--	--	100	--
- Fraiseuse à contrôle numérique	--	--	100	--
- Tour multi-têtes à chargement rotatif	--	--	--	300
- Perçeuse spécialisée et dédiée	--	--	--	500
- Presse manuelle d'injection d'époxie	2	2	--	--
- Presse automatique à injection multiple	--	--	20	20
- Outil et autres	5	5	30	60
A - TOTAL:	62	62	400	880
Installation et mise en marche: 5% de A	3.1	3.1	20	44
Coût total d'installation d'une ligne de production	65.1	65.1	420	924

TABLEAU III - Description des composantes de l'option 2

COMPOSANTES ET ETAPES DE FABRICATION	TEMPS DE REALISATION POUR LES PRODUCTIONS SUIVANTES (minutes/pièce)			
	1	10	10,000	100,000
1. Corps du support (Figure 2.1)				
- Dégrossissage et tournage des dia. extérieurs	120	60	10	2
- Perçage des cavités pour l'insertion du mécanisme interne	5000	2500	60	12
2. Embout (Figure 2.2)				
- Dégrossissage et tournage de la pièce	60	30	5	2
- Perçage de la cavité	60	30	5	2
- Traitement thermique (55 à 60 Rc)	30	15	1	0.5
- Meulage intérieur et extérieur de la pièce	400	200	15	5
- Perçage des orifices	120	60	5	1
3. Valve (Figure 2.3)				
- Dégrossissage des diamètres extérieurs	60	30	3	1
- Traitement thermique	30	15	1	0.5
- Meulage des diamètres extérieurs	500	250	15	5
4. Guide de valve (Figure 2.4)				
- Tournage et filetage de la pièce	60	30	5	2
- Perçage et alésage du diamètre intérieur	100	50	5	2
- Rectifier le diamètres intérieur par le procédé de "ballizing"	400	200	15	5
- Traitement thermique (55 à 60 Rc)	30	15	1	0.5

TABLEAU III (suite)

COMPOSANTES ET ETAPES DE FABRICATION	TEMPS DE REALISATION POUR LES PRODUCTIONS SUIVANTES (minutes/pièce)			
	1	10	10,000	100,000
5. Siège du ressort (Figure 2.5)				
- Tournage	20	10	2	0.5
- Traitement thermique	20	10	0.5	0.5
- Rectifiage	30	15	3	1
6. Vis d'ajustement de la pression (Figure 2.6)				
- Tournage et filetage	120	60	6	2
- Usinage des gorges	60	30	1	0.5
7. Vis d'ajustement de l'ouverture (Figure 2.7)				
- Tournage et filetage de la vis	60	30	6	2
- Fraisage des méplats	15	8	1	0.5
- Traitement thermique	20	10	1	0.5
- Meulage	120	60	5	2
8. Tuyau d'arrivée de carburant (Figure 2.8)				
- Tournage du diamètre de jonction	10	5	2	0.5
- Perçage de la cavité pour insertion du filtre	20	10	3	1
- Formage de l'évasement du bout du tuyau	20	10	2	1
- Cintrage du tuyau	10	5	1	0.5
9. Filtre				
- Tournage	40	20	3	1
- Traitement thermique	20	10	1	0.5
- Fraisage	60	30	3	1

TABLEAU III (suite)

COMPOSANTES ET ETAPES DE FABRICATION	TEMPS DE REALISATION POUR LES PRODUCTIONS SUIVANTES (minutes/pièce)			
	1	10	10,000	100,000
10. Connecteur (Figure 2.10)				
- Tournage	80	40	5	2
- Traitement thermique	30	15	1	0.5
- Meulage	60	30	6	2
11. Assemblage				
- Tuyau d'alimentation au corps par une brazure au four	200	100	10	5
- Embout au corps par une brazure au four	200	100	10	5
- Guide au corps à l'aide d'époxie	500	250	15	6
- Vérification des valves avec chacun des embouts	60	30	3	1
- Assemblage final	15	8	5	2
A - TOTAL:	8760 =====	4381 =====	244.5 =====	79 =====
- Manutention et essais: 10% de A	840	438	24.5	79
- Imprévus: 10% de A	840 -----	438 -----	24.5 -----	79 -----
B - TOTAL:	10080 =====	5257 =====	293.5 =====	94.8 =====

TABLEAU III (suite)

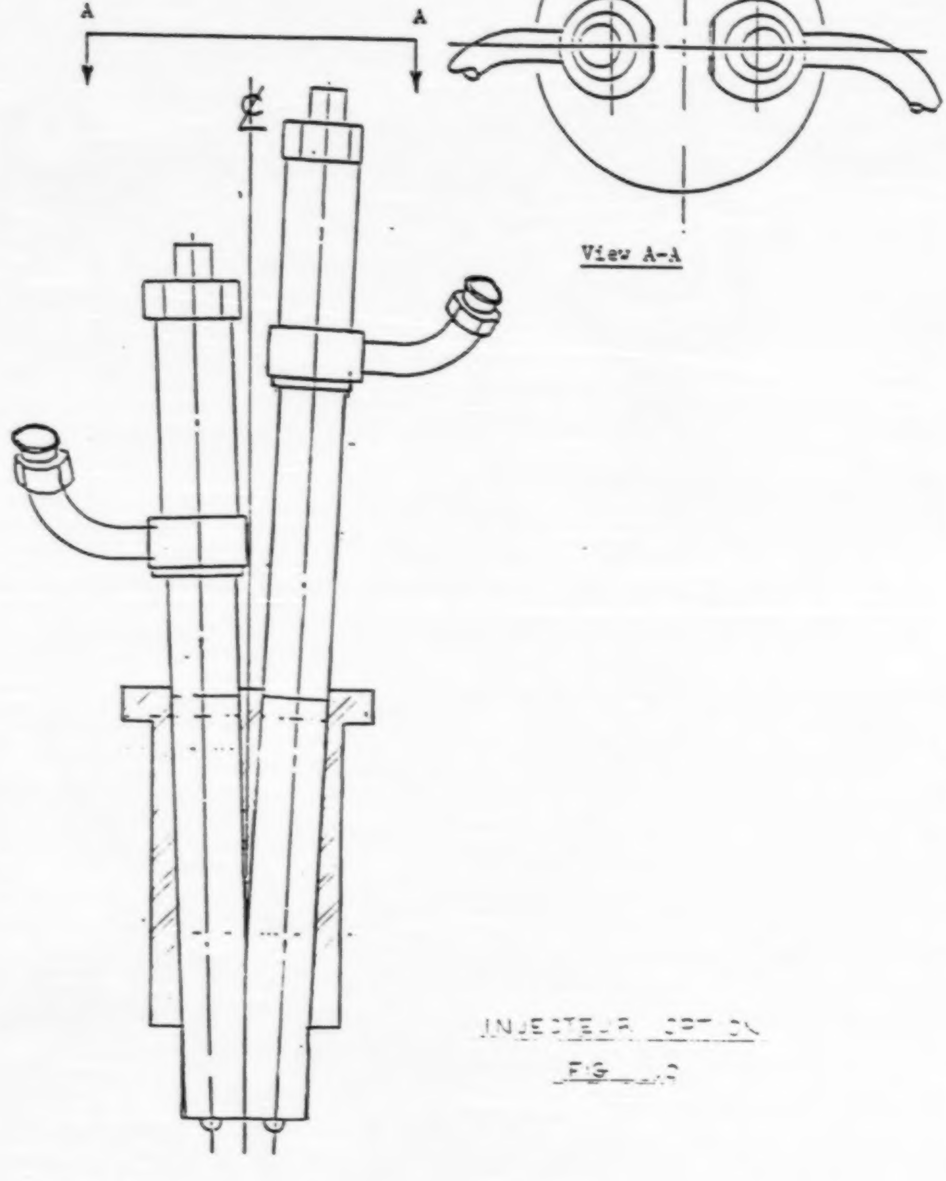
COMPOSANTES ET ETAPES DE FABRICATION	TEMPS DE REALISATION POUR LES PRODUCTIONS SUIVANTES (minutes/pièce)			
	1	10	10,000	100,000
- Coût de main d'oeuvre: $40. X B/60	6720	3504	196	63.20
- Matériel: - Ecrous, époxie, ressort, acier, etc.	20	10	5	2
Coût total unitaire (sans amortissement de l'équipement)	6740 =====	3514 =====	201 =====	65.20 =====
Taux de production d'une ligne (pièce/heure):	0.007	0.015	1	5

TABLEAU IV - Coûts des équipements prévus pour l'option 2

DESCRIPTION	COUT DES EQUIPEMENTS ($K)			
	1	10	10,000	100,000
Tour standard de très bonne précision	30	30	--	--
Rectifieuse portative pour monter sur un tour	3	3	--	--
Rectifieuse cylindrique universelle à commande numérique	--	--	100	--
Tour à contrôle numérique	--	--	150	--
Fraiseuse à contrôle numérique	100	100	--	--
Centre d'usinage robotisé	--	--	350	--
Appareil d'usinage par électro-érosion (EDM) ou perceuse à commande numérique	100	100	100	--
Four à traitement thermique et brazure	60	60	100	--
Lignes automatisées pour traitement thermique	--	--	--	250
Ligne automatisée pour brazure au four	--	--	--	250
Presse automatique pour effectuer le "ballizing"	--	--	50	50
Cintreuse automatisée	--	--	30	30

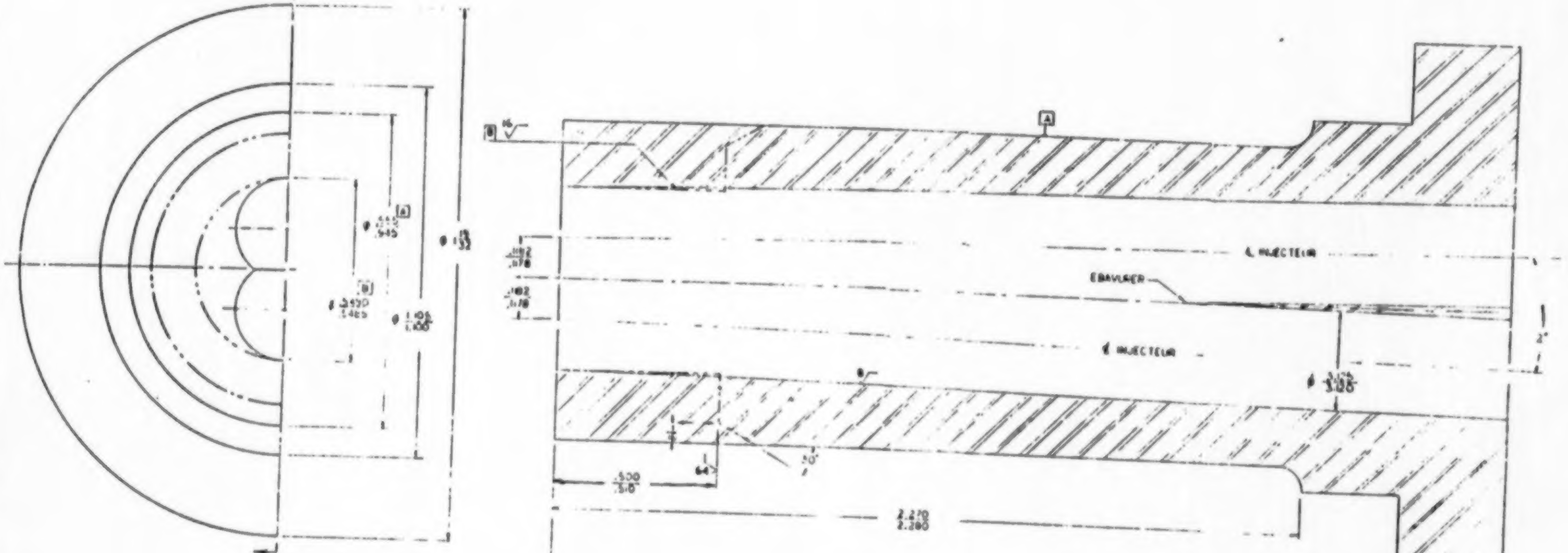
TABLEAU IV (suite)

DESCRIPTION	COUT DES EQUIPEMENTS ($K)			
	1	10	10,000	100,000
Cintreuse manuelle	5	5	--	--
Presse mécanique à double action	20	20	--	--
Presse à "swaging"	--	--	50	50
Tour spécialisé pour fabrication de vis (screw machine)	--	--	--	300
Appareil d'usinage par électro-érosion à multiples points de sortie	--	--	--	200
Tour multi-têtes à chargement rotatif	--	--	--	300
Appareils dédiés aux perçages des ouvertures	--	--	--	500
Outils, montages et autres	10	15	50	100
A - TOTAL	328	333	980	2030
Installation et mise en marche: 5% de A	16.4	16.7	49	101.5
Coût total d'installation d'une ligne de production	344.4	349.7	1029	2131.5



INJECTEUR OPTION

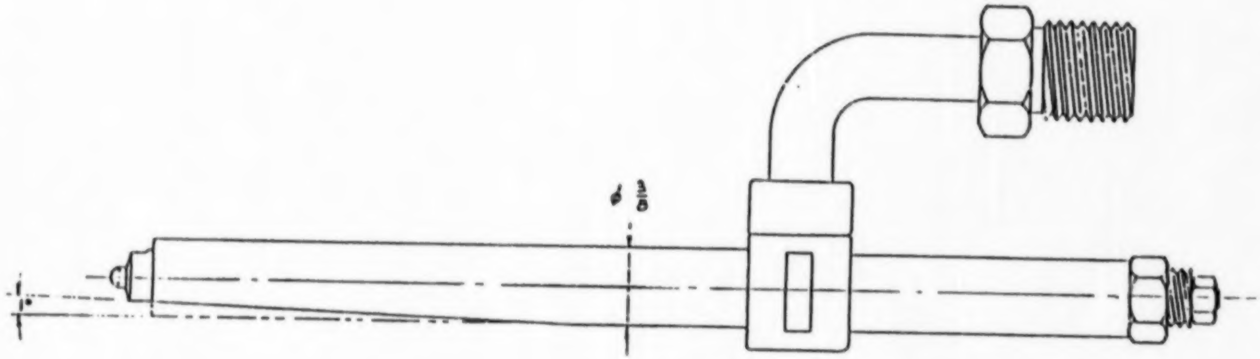
FIG [illegible]


A
B
16
Ø .555 / .545
Ø 1 3/32
Ø .3490 / .3485
Ø 1.105 / 1.100
.182 / .178
.182 / .178
¢ INJECTEUR
EBAVURER
¢ INJECTEUR
Ø 3.125 / 3.120
2°
30°
1/64
.500 / .510
2.270 / 2.280
2 7/8
5/16

LA SECTION REPRESENTEE PAR UN TRAIT INTERROMPU SERA USINEE APRES L'ASSEMBLAGE DES INJECTEURS.
LES INJECTEURS SERONT ASSEMBLES EN UTILISANT UN ADHESIF.
LES ¢ SERONT REFERENCES PAR RAPPORT AU Ø A

FIG 1.1

SUPPORT D'INJECTEUR
CONCEPT NO. 3
4140
iVi INC.
6511801



N.B. LE TRAIT INTERROMPU REPRESENTE L'INJECTEUR DANS SON ETAT ORIGINAL.

LES INJECTEURS UTILISES SERONT DES STANADYNE NO 20502 ET 22202.

FIG 1.2.

INJECTEUR OPTION·

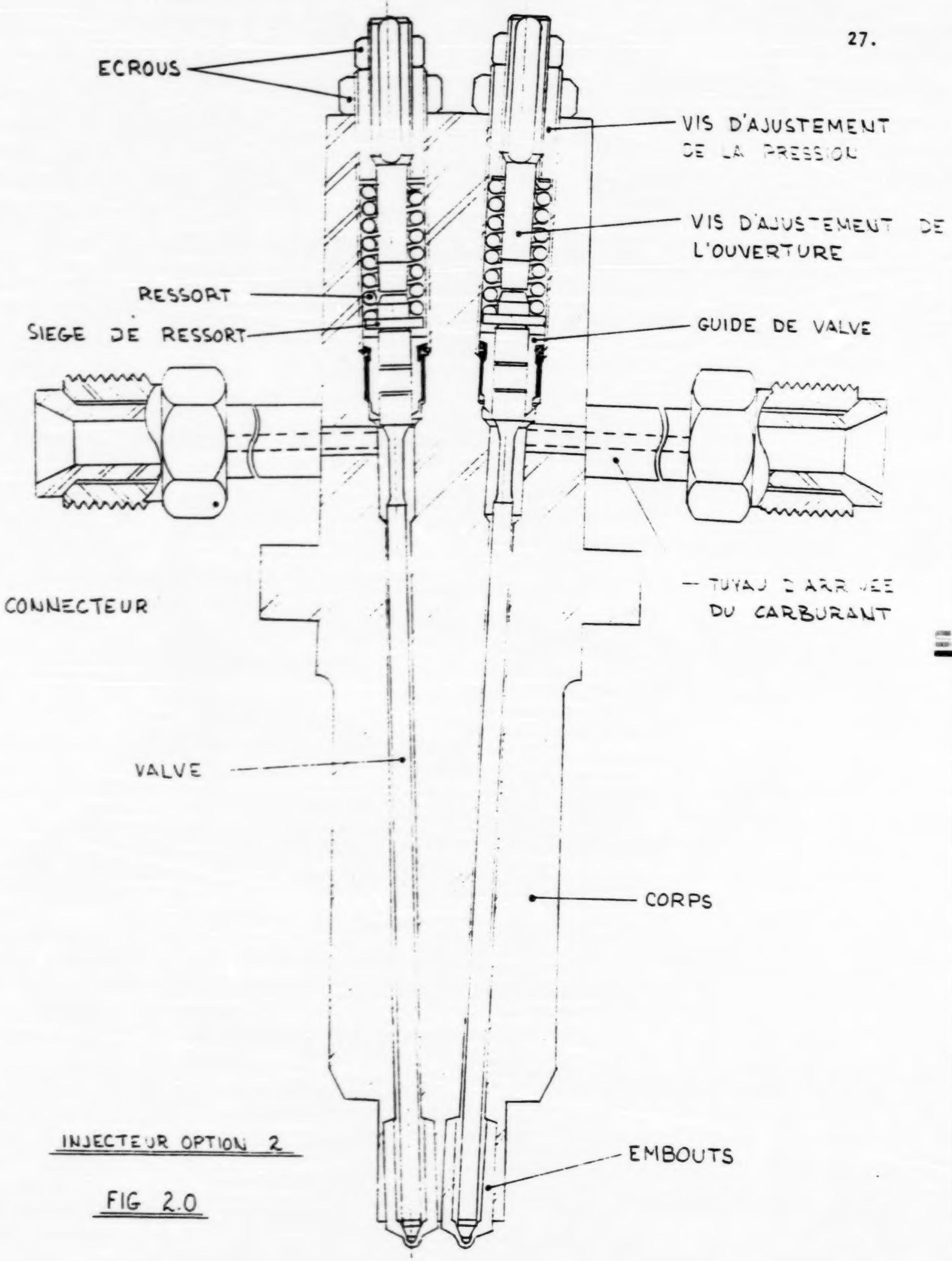
WAUKESHA CFR 48

2

06-11-85

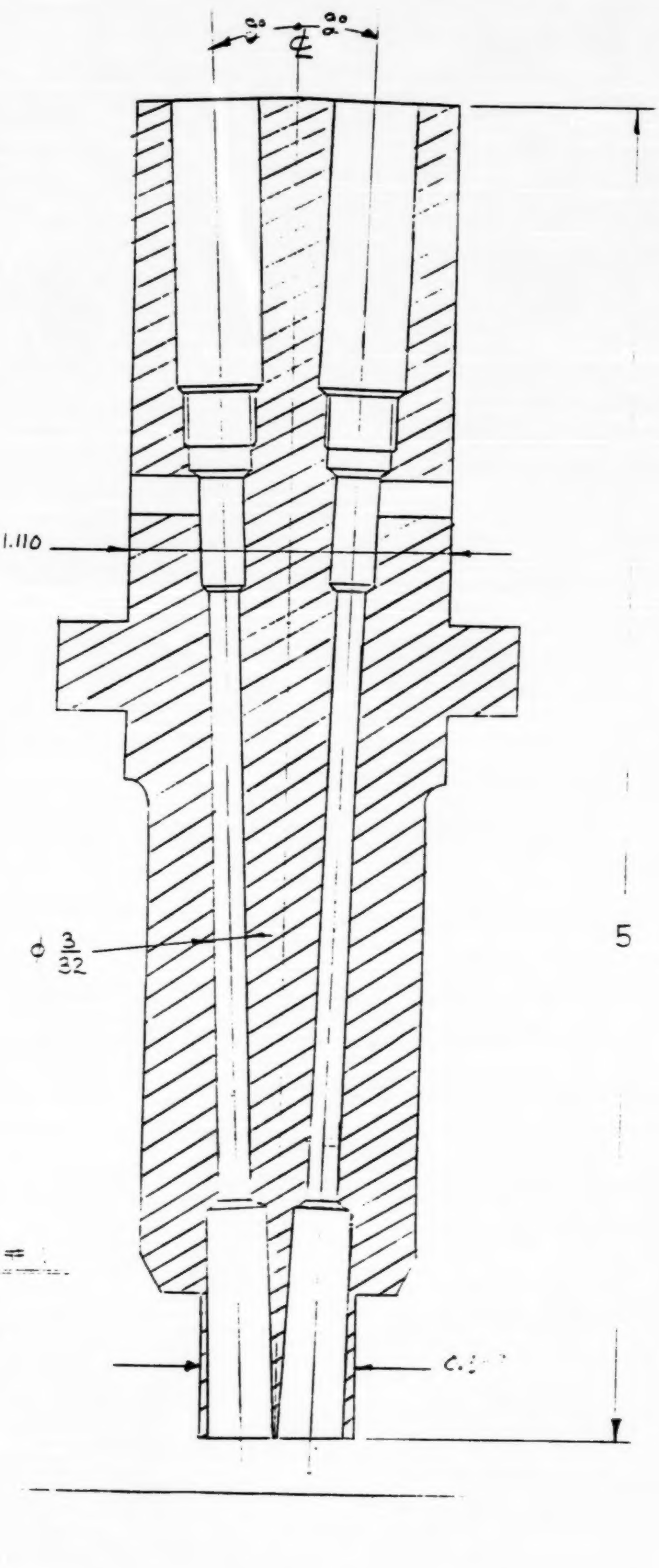
2

LES INDUSTRIES VALCARTIER INC


ECROUS
VIS D'AJUSTEMENT DE LA PRESSION
VIS D'AJUSTEMENT DE L'OUVERTURE
RESSORT
SIEGE DE RESSORT
GUIDE DE VALVE
CONNECTEUR
TUYAU D'ARRIVEE DU CARBURANT
VALVE
CORPS
EMBOUTS

INJECTEUR OPTION 2

FIG 2.0


2°
℄
2°
ϕ 1.110
ϕ 3/32
5
0.5

PROBE OPTION # 1

FIG 2.1

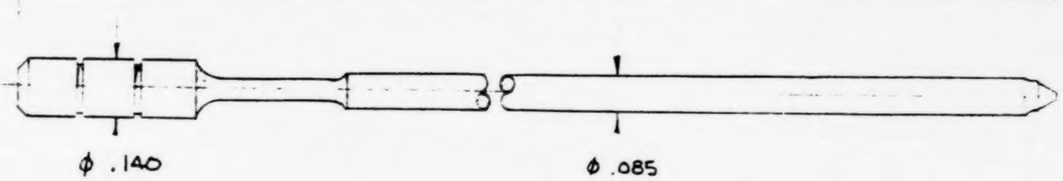


FIG 2.3 : VALVE

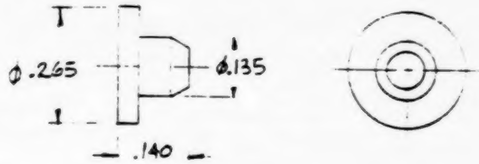

ϕ .265
ϕ .135
.140

FIG 2.5 : SIEGE DE RESSORT

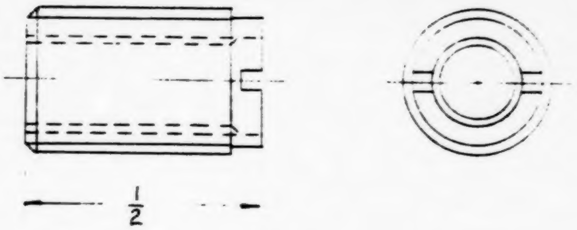


FIG 2.6 : VIS D'AJUSTEMENT DE LA PRESSION

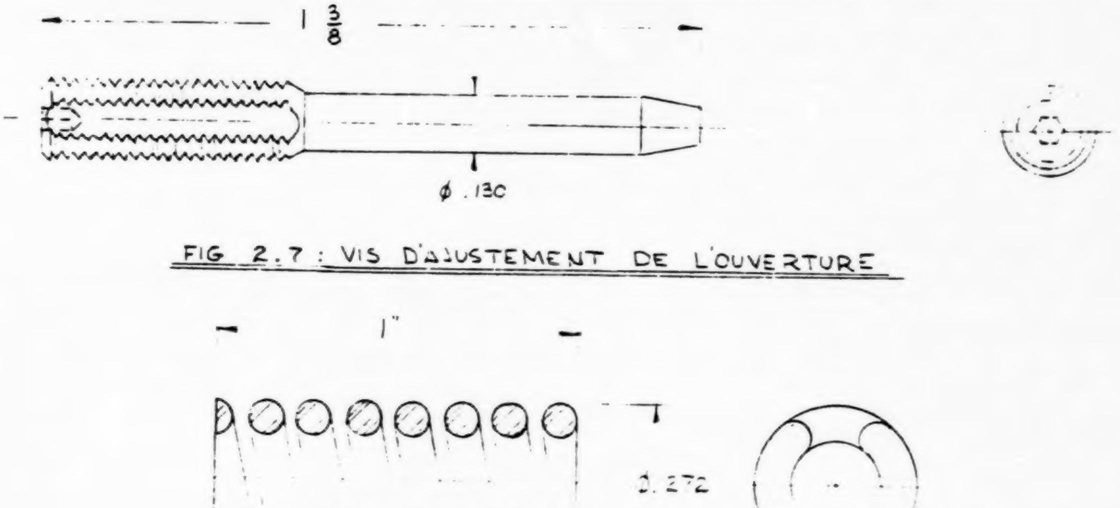


FIG 2.7 : VIS D'AJUSTEMENT DE L'OUVERTURE

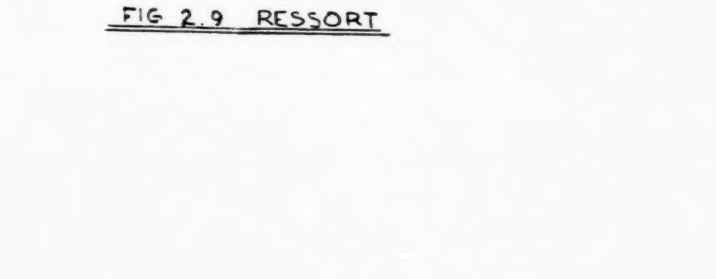


FIG 2.9 RESSORT

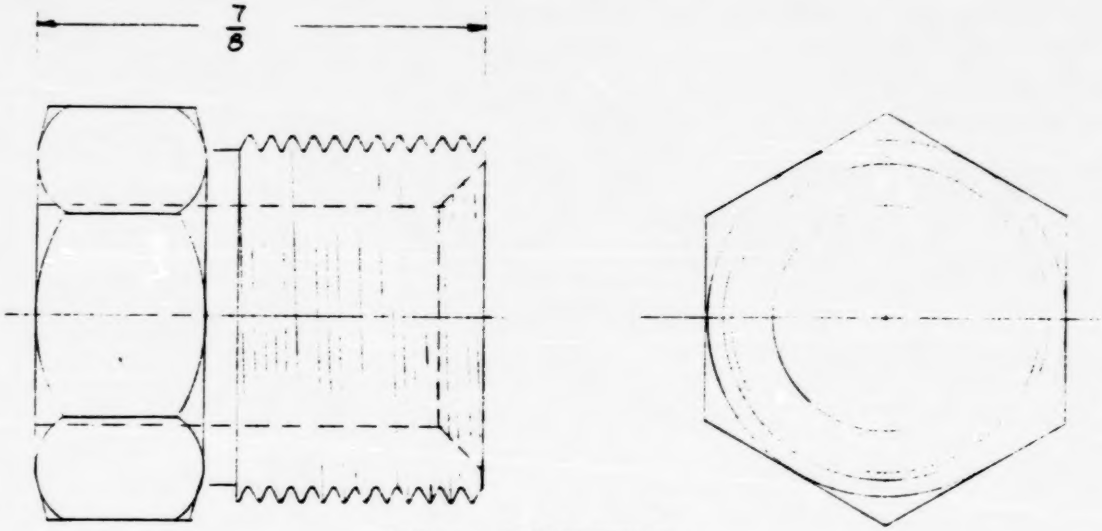


FIG 2.10 : CONNECTEUR

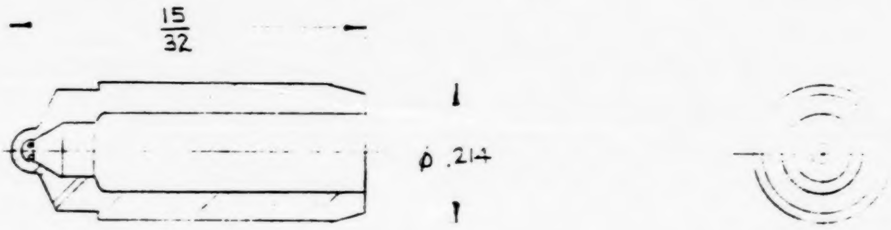


FIG 2.2: EMBOUT

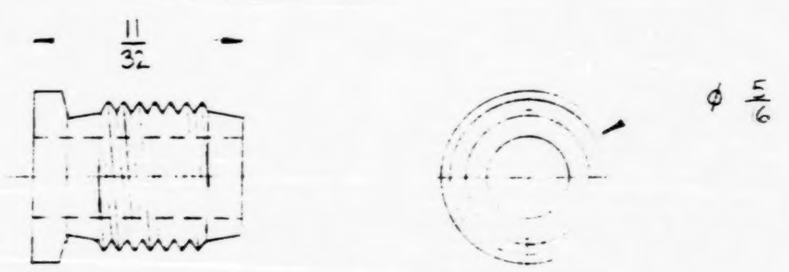


FIG 2.4 : GUIDE DE VALVE

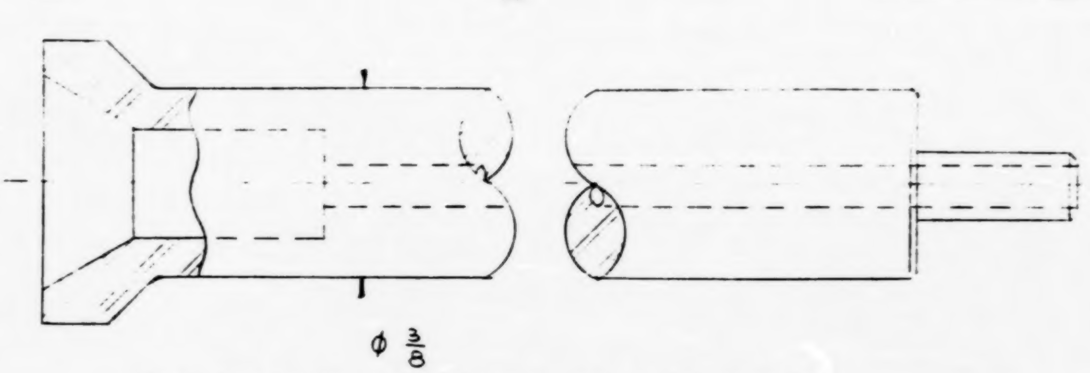


FIG 2.8 : TUYAU D'ARRIVEE DU CARBURANT